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APPLICATION OF A POWDER LUBRICATION SYSTEM TO A GAS TURBINE ENGINE

PART III

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Air Force Aero Propulsion Laboratory

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(Prepared under contract No. AF33(615)-1331 by the Stratos Division, Fairchild Hiller Corporation, Bay Shore, New York; Stanley Wallerstein, author).

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Abstract

The feasibility of adapting a powder lubrication system for the lubrication of the main shaft bearings of an unfired J-69 gas turbine engine was established, in lieu of the oil lubrication system used in the standard engine. Ball and roller bearings have been successfully lubricated with powder-lubrication in a J-69 unfired gas turbine engine. Ball bearings have been successfully operated for periods of 35 hours with powder lubricants. Thrust load tests to 500 pounds at 8000 rpm were conducted. Testing with a 100 pound thrust load to 20,000 rpm with heated (bleed air temperature) carrier air was successfully performed on two different bearing designs. Both bearings were deep groove, split inner ring design with a one piece outer ring guided retainer made of silverplated AMS6415 steel; one bearing had inner and outer races and balls of M-50 vacuum melt tool steel; the other had the inner and outer races and balls of 440CM modified corrosion resistant steel. Roller bearings have performed successfully through a speed range of 8,000 to 20,000 rpm at stabilized operating temperatures of 515°F, simulating estimated operating conditions. The bearing had inner ring guided rollers and an outer ring guided retainer. Rings and rollers were of M-50 vacuum melt tool steel and the retainer was Monel S.

The lubricant used for all the evaluations was a powder mixture consisting of five parts of micronized Acheson No. 38 graphite plus one part of laboratory-grade cadmium oxide entrained in an air carrier. The optimum lubricant powder flow for ball bearing operation under thrust loaded conditions is about 0.015 grams per minute.

Redesign and retrofit of a J-69 gas turbine engine with a powder lubrication system was completed so that engine could be tested under fired conditions.

APPLICATION OF A POWDER LUBRICATION SYSTEM TO A GAS TURBINE ENGINE

PART III

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Foreword

This report was prepared at Stratos Division, Fairchild Hiller Corporation, Bay Shore, Long Island, New York under USAF Contract No. AF33(615)-1331. This contract was initiated under Project No. 3048, Task No. 304806 and presents the results of work performed during the period from June 1966 to June 1967. The manuscript of this report was released by the author October 1967 for publication as an AFAPL Technical Report.

The work was administered under the Air Force Aero Propulsion Laboratory, Directorate of Laboratories, Air Force Systems Command, Wright-Patterson Air Force Base, Ohio with Mr. G.A. Beane IV and Mr. J.B. Schrand (APFL), Project Engineers.

The author wishes to acknowledge the contributions to the program of Mr. Wallace Mende. The assistance of Mr. Joseph Arnold in the conduct of the tests in the Stratos Division Laboratories, Supervised by Mr. George Jenney, is also acknowledged.

This technical report has been reviewed and is approved.

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SECTION I

INTRODUCTION

Powder lubrication is defined as lubrication by a solid film that is continuously being deposited on a surface from a suspension of solid particles in a carrier gas. The performance of this lubrication technique is a function of both the carrier gas and the lubricant. The important parameters* are as follows:

- Ratio of carrier gas to lubricant
- Velocity of the gas stream
- Mixture ratio of the lubricant
  (Relative proportions by weight of the powder components)
- Particle size
- Thermal stability
- Filming qualities of the lubricant

Prior research programs have also investigated the endurance capability of bearings and gears lubricated with powder lubricants while operating at temperatures to 1200° F. These studies examined the following specific areas:

- Bearing and gear material
- Bearing and gear design
- Lubricants
- Method of lubricant application

The evidence obtained from prior investigations shows that the powder lubrication technique is capable of lubricating bearings and gears at high temperatures and is feasible for lubricating a turbine engine. The major problem areas that have been investigated under this program are as follows:

- Lubricant efficiency
- Heat rejection requirements
- Bearing design
- Growth and distortion due to thermal gradients

*These parameters have been investigated in previous programs (Reference 1 and 2.)

- Powder lubrication distribution system

  a. Lubricant supply
  b. Lubricant exhaust
  c. Seals

- Materials

The lubricant selected for use in this program is a mixture of five parts by weight of graphite to one part of cadmium oxide. This lubricant was selected on the basis of its ability to provide a lubricating film at temperatures ranging from ambient room temperature to temperatures above 800° F. The powder lubricant can be used below room temperature, providing that at temperatures below 32° Fahrenheit both the powder and carrier air are absolutely dry and free of water moisture. Otherwise, the powder would tend to cake and solidify. A more detailed discussion of lubricant selection is presented in section 5. Background information, which includes the analysis of adaption of the J-69 Turbine engine for powder lubrication, is presented in reference 3.

The complexity of the turbine engine required that preliminary testing be accomplished on a separate bearing test rig prior to testing the powder lubricated bearings in an actual J-69 engine. This test rig was designed to measure the powder lubricant efficiency as compared to the efficiency of the existing engine lubrication system. This information enabled the engine bearing temperatures to be predicted when being lubricated with powder lubricant. The results of these tests as well as the engine thermal analysis is presented in reference 4.

The next phase of the program was to install a powder lubrication system in a J-69 engine and test the engine at moderate speeds, loads and temperatures to optimize the bearing design and powder lubricant supply and exhaust systems. This phase of testing has been previously reported in reference 4.

This report deals primarily with operation of powder lubricated bearings in an unfired J-69 engine at conditions of speed, load, and temperature simulating those expected in an actual fired engine. The report discusses work performed from June 1966 to June 1967 with respect to the following:

- Unfired J-69 engine testing at simulated fired engine conditions of speed, load and temperature.

- Finalized design of powder lubrication system for use with a fired J-69 engine.

- Modification of fired J-69 engine for use with powder lubrication.

SECTION II

SUMMARY

The third year of this program was primarily devoted to the testing of an unfired, powder lubricated J-69 engine at speeds up to 20,000 RPM at bearing temperatures and loads which simulated those expected in a fired engine.

Earlier engine testing (Reference 4) was concerned with selection of bearing materials and development of bearing design for use in the engine. Bearing materials were selected primarily on the basis of estimated operating temperatures, in conjunction with good lubricant filming characteristics. Low speed testing was successfully performed during the earlier report period at 8,000 and 12,000 RPM with the engine rear (roller) bearing heated to 540° F.

Testing during this phase of the program was concerned primarily with operation of the unfired engine at 20,000 RPM with the rear bearing operating at a temperature of 540° F, and with 50 to 100 pounds thrust loading on the front bearing. A roller bearing was successfully operated under design conditions and ball bearings of two different designs were successfully operated under design conditions of speed and load.

A complete design review of the conventionally lubricated J-69 engine was undertaken, and a powder lubrication manifolding and seal system was designed to be incorporated into the engine with a minimum number of changes to the existing engine components.

A second J-69 engine was disassembled, modified with the incorporation of a powder lubrication system, and reassembled for final testing of Wright-Patterson Air Force Base, Dayton, Ohio.

SECTION III

DISCUSSION

GENERAL

USAF Contract No. AF33(615)-1331 has as its objective the design and fabrication of a lubrication system for a J-69 turbojet engine utilizing powder lubrication in lieu of conventional oil lubrication of the engine rotor shaft main bearings. Originally the Pratt and Whitney J-57 gas turbine engine was considered for this program, but as this engine was very large and complex; it was mutually agreed that a smaller, simpler engine would be selected. Three candidate engines were evaluated; the J-69 engine manufactured by Continental Aviation and Engineering Corporation; the T-53 engine manufactured by the Lycoming Division of AVCO Corporation, and the T-63 engine manufactured by the Allison Division of the General Motors Corporation. Of the three engines the J-69 was selected for the program as both drawings and engines were available. The T-53 and T-63 were both eliminated because of the reduction gear boxes which contained planetary gear sets, and were newer engines and not considered as reliable as the J-69.

However, before a powder lubrication system could be incorporated into an operating engine, studies and testing were needed, both of system components (lubricant distribution system and powder lubricated bearing design) and of the entire engine lubricating system. The scope of this system covers the entire engine, less accessory gear train and gearbox, and is divided into the following study areas:

- Seals and main engine bearings
- Powder dissemination
- Manifold and plumbing
- Engine hardware
- Engine test program

TEST ENGINE DESIGN AND PRINCIPAL PERFORMANCE CRITERIA

The turbojet engine used as the test engine for this program is a USAF Model J69-T-25 manufactured by the Continental Aviation and Engineering Corporation (Figure 1). It has a normal rated thrust of 880 pounds with a minimum thrust of 1025 pounds for takeoff. Engine performance ratings are listed in Table 1. The engine has a maximum diameter of 22.3 inches, a length of approximately 50 inches (not including tail pipe extension) and a dry weight of 364 pounds. The engine is of the typical turbojet type, with the exception of the centrifugal fuel distributor, which is built into the main rotor shaft.

The air enters through a three-strut-supported intake housing and is compressed by a centrifugal impeller into an annular combustion chamber. Combustion products

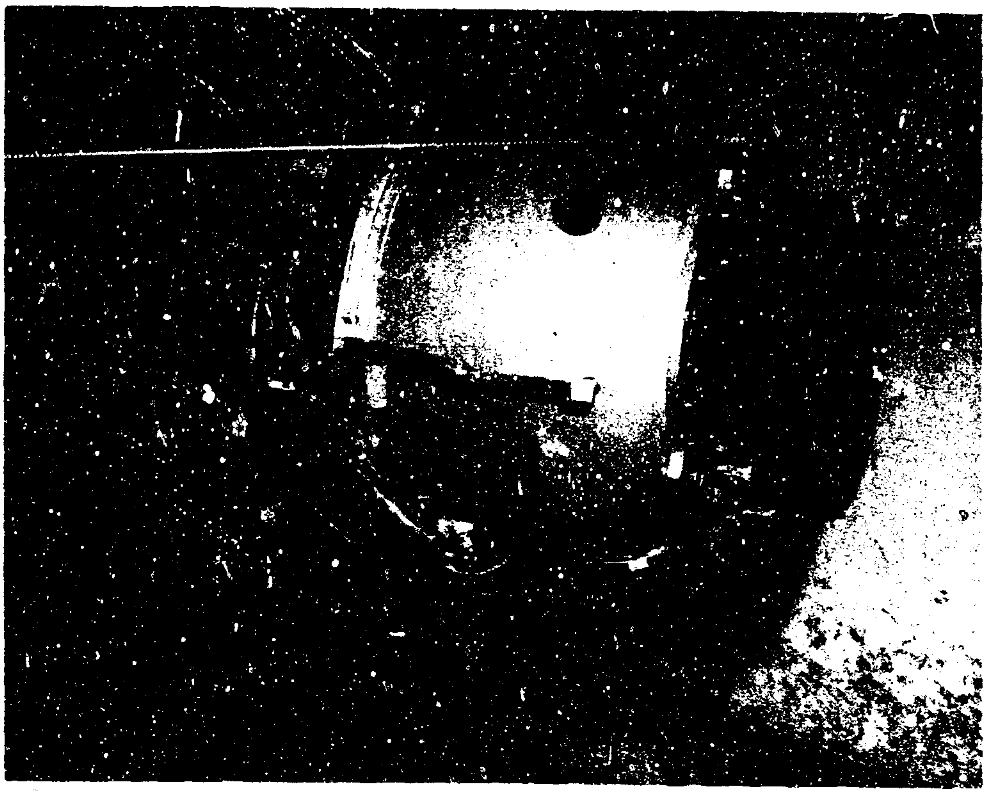


Figure 1. Test Engine

TABLE 1.
ENGINE PERFORMANCE

Performance Ratings	*Engine Speed		*Minimum Thrust (Lb)
	RPM	Percent	
Maximum	21730	100	1025
Military	21730	100	1025
Normal	20700	95	880
90% Normal	20000	92	795
75% Normal	19000	88	660
Idle	7820-8260	36-38	70 Max.

*Standard day conditions of 29.2 in Hg and 59° F and secondary air are directed by a stator assembly into a single-spool, fir-tree footed turbine wheel. The tail cone is abbreviated and is usually designed to suit particular airframe assemblies.

This program deals with the lubrication of the main rotor shaft and its bearings. The complete shaft assembly consists of the compressor, fuel distributor, thin walled spacer, turbine wheel, and outboard stub shaft and is supported by two identical-bore (35 mm) bearings. The front main bearing is a deep-race ball bearing and takes up thrust while the aft bearing is a cylindrical roller bearing that permits a limited axial growth of the shaft due to operating thermal conditions.

The primary purpose of this program was to demonstrate the feasibility of using powder lubricated bearings in an application such as a turbojet rotor shaft. The reason for using a powder lubricant is to provide the adequate lubricant performance in a component (bearing or gear) normally operating in a high-temperature environment. In a moderate-temperature environment, such as the J-69 rotor bearings, the substitution of powder for the normal oil lubrication system actually causes an increase in bearing operating temperature, as the cooling properties of powder lubricant and powder carrier air are very small compared with those of oil lubricants. At the J-69 rear bearings, actual operating temperature with oil lubrication is about 330° F, while the calculated temperature using powder lubrication is 530° F. This temperature difference should only affect the areas immediately adjacent to the rear bearing and the change from oil to powder lubrication was not expected to have any effect on engine performance other than that concerned with bearing operation.

The primary task of this program was to evaluate the use of powder lubricants on the bearings and seals at engine operating conditions using engine bleed air as the carrier gas. Typical test requirements are listed in table II.

TABLE II
TEST REQUIREMENTS

Item	Limit	Remarks
Allowable Speed Fluctuation		
8040 RPM (37 PCT)	±50 RPM (0.23 PCT)	
21730 RPM (100 PCT)	±25 RPM (0.12 PCT)	
Vibration (at front and rear)	1.5 mils	Maximum Above 16,000 RPM
	3.5 mils	Maximum Below 16,000 RPM
	5.0 mils	Transient Peak between 10,000 and 16,000 RPM

POWER EXTRACTION

Accessory	Torque, Lb-In.	HP	Speed, RPM
Starter-Generator	100	12.4	7,823
Oil-Pump	13	1.0	4,694
Tachometer	7	0.5	4,224
Fuel Control	9	0.5	3,663
Fuel Pump	69	4.0	3,663
Hydraulic Pump*	175	10.2	3,663
Main Turbine Shaft		28.6	21,730
*Optional for this test setup.			

STARTER - Minimum Speed and Torque

Condition	Torque Lb-Ft 60° F	Starter Drive RPM	Engine Rotor RPM
Firing	12	432	1200
Cut-out	8	1800	5000

SECTION IV

TEST BEARINGS

GENERAL

One roller bearing and one ball bearing are used to support the engine main rotor assembly. Both bearings are of the basic 207 size (light series, 35-mm bore) with modifications for the high rotational speeds encountered in this application. General characteristics of the bearings used both in bearing rig and in engine testing are described in the following paragraphs. Inner and outer race curvature of all the ball bearings are as follows:

disc radius = 52 percent of ball diameter, or
disc diameter = 104 percent of ball diameter.

Detail modifications are listed in the test summary tables.

BALL BEARING SERIES B-1 (Actual Engine Bearing)

The engine front main bearing is a ball bearing of split-inner-ring, deep-groove construction to accommodate high thrust loads, as shown in figure 2.

The production bearing, as supplied with the engine, has 12 balls of 7/16-inch diameter. Balls and rings are of 52100 bearing steel. The one-piece retainer is silver plated, forged silicon-iron bronze (AMS4616) and is of outer-ring guided design.

BALL BEARING SERIES B-2

Bearing B-2, shown in figure 3, is similar to B-1, but with rings and balls of M-50 vacuum-melt tool steel. The one-piece inner-ring guided retainer is silver-plated AMS 6415 steel.

BALL BEARING SERIES B-3

Bearing B-3 is basically the same as B-1, with the exception that the balls and rings are of M-50 vacuum-melt tool steel.

BALL BEARING SERIES B-4

Bearing B-4, shown in figure 4, is of deep groove, split inner ring design with a special X-shaped retainer. Rings and the 12 balls (7/16-inch diameter) are of M-50 vacuum-melt tool steel. The retainer, which was designed for good lubricant flow-through characteristics, is of Monel S.

BALL BEARING SERIES B-5

Bearing B-5, as shown in figure 5 is of deep-groove split-inner ring design with balls and rings of 440 CM modified stainless steel. The one-piece outer ring guided retainer is of silver plated AMS-6415 steel.

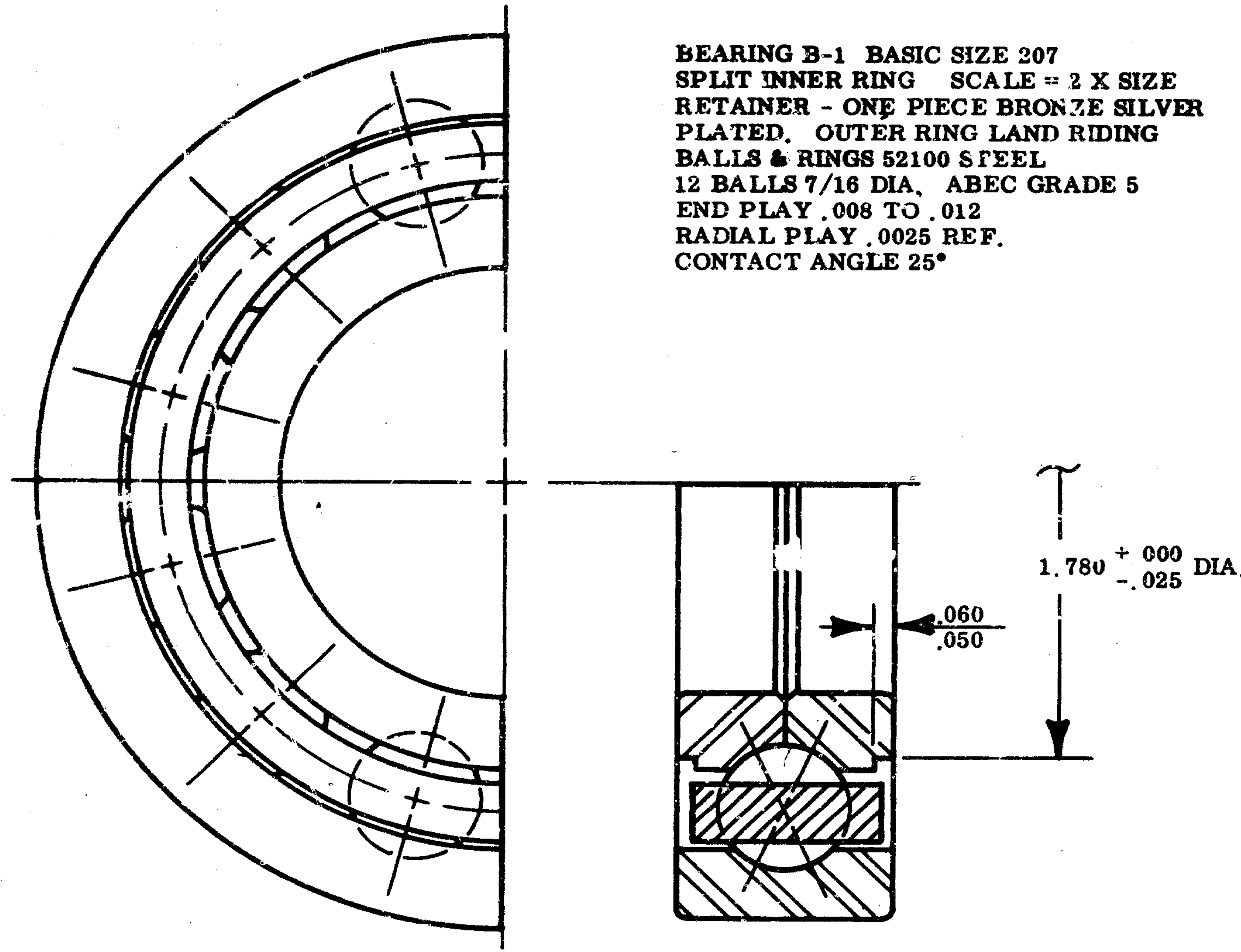


Figure 2. Test Bearing B-1

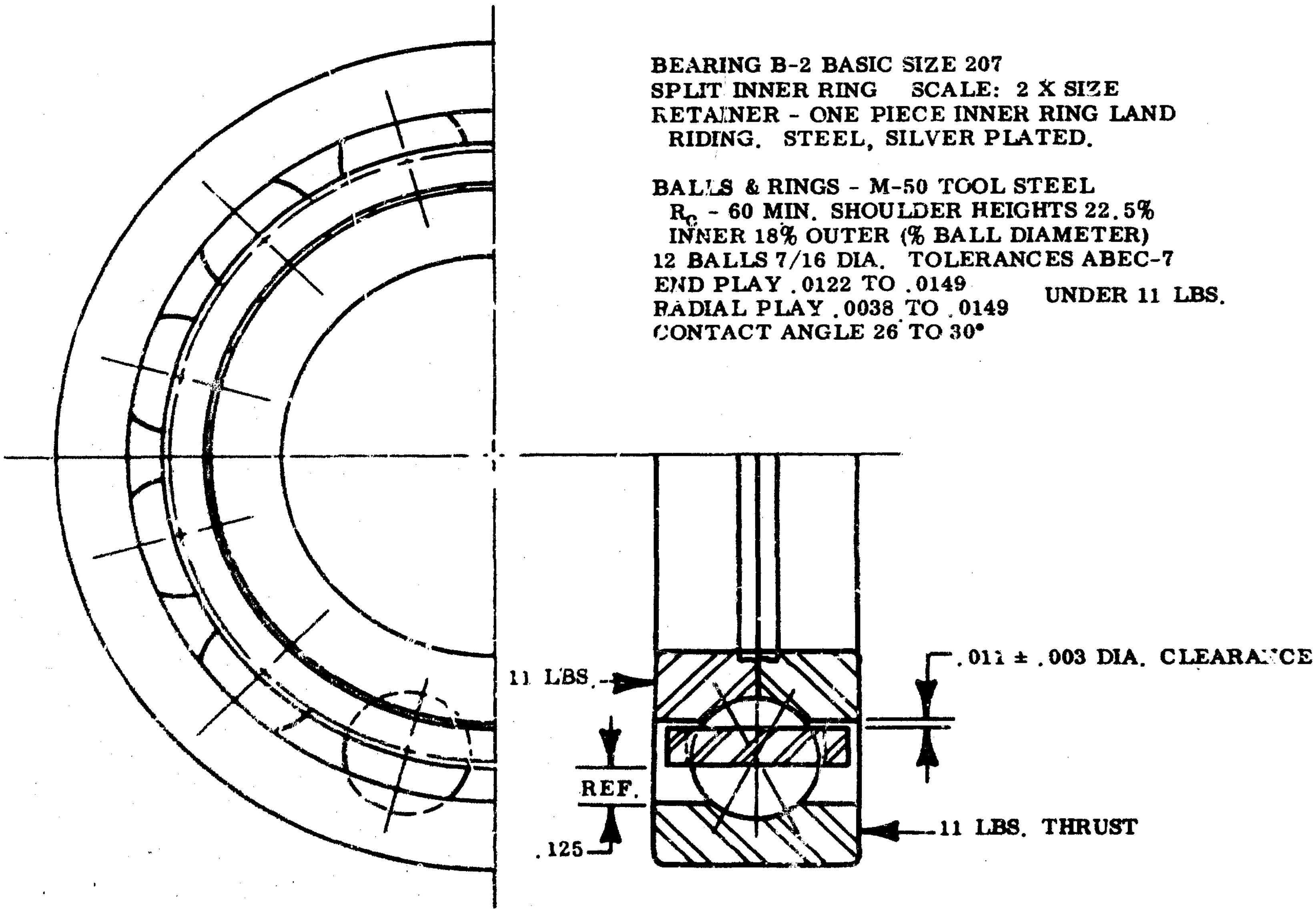


Figure 3. Test Bearing B-2

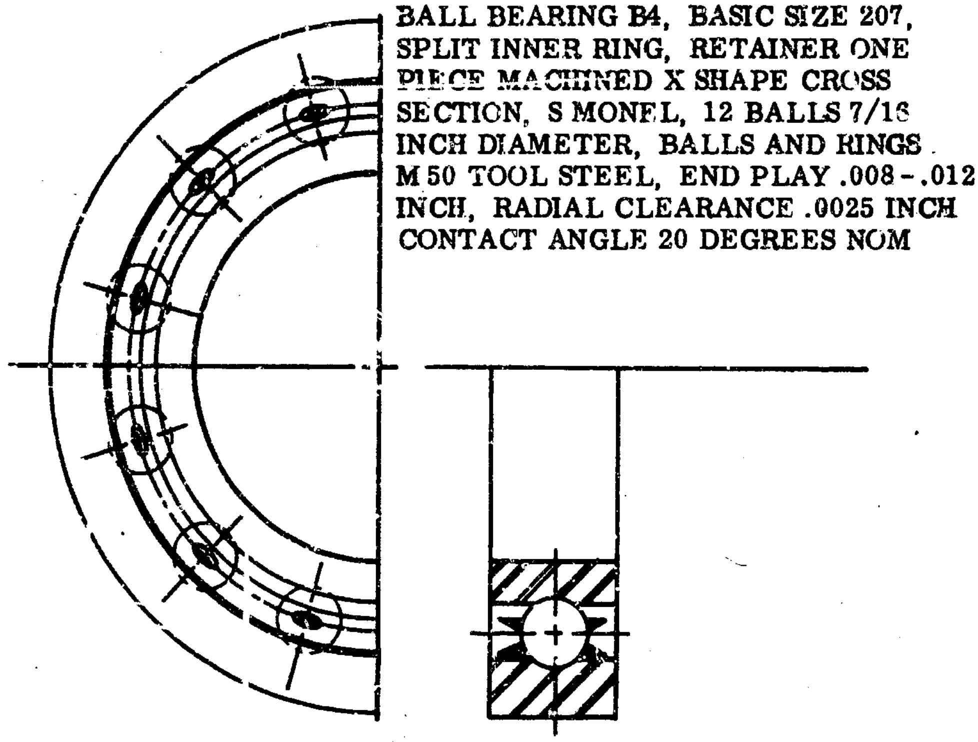


Figure 4. Test Bearing B-4

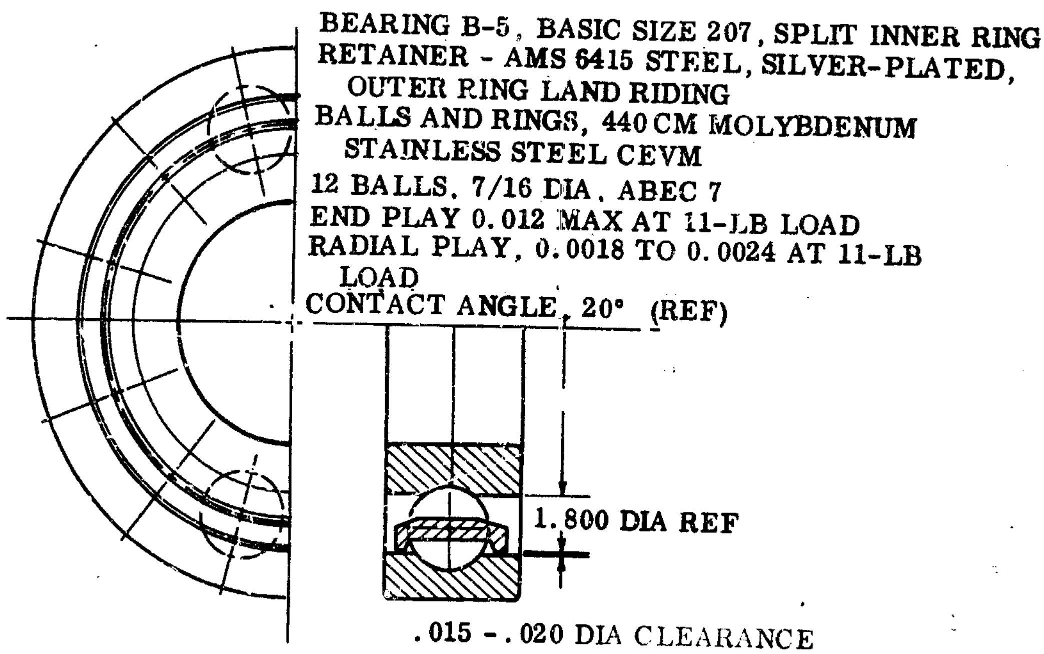


Figure 5. Test Bearing B-5

ROLLER BEARING SERIES R-1 (Actual Engine Bearing)

The conventional engine rear main bearing is a cylindrical roller bearing with a straight-through inner ring and two-lip outer ring for roller guidance. Twelve rollers, 11/32-inch diameter by 11/32-inch long, are used. Rollers and rings are 52100 bearing steel, and the retainer is silver-plated extruded brass and is of two-piece riveted design. The retainer is roller guided, as shown in figure 6.

ROLLER BEARING SERIES R-2

Bearing R-2, shown in figure 7, has a straight-through outer ring and two-lip inner ring for guidance of the 12 rollers, which are 11/32-inch diameter and 11/32-inch long. Rings and rollers are 52100 bearing steel. The retainer is of two-piece separable design, riveted and roller guided, and is made of silver-plated brass.

To improve lubricant flow through the bearing, and to prevent caking of powder in the inner race, eight grooves were machined in the inner-ring lip on the lubricant exhaust side of the bearing, as shown in figure 7.

ROLLER BEARING SERIES R-3

Bearing R-3, shown in figure 8, has a straight-through outer ring, a two-lip inner ring for roller guidance, and 14 rollers, each 10-mm diameter by 10-mm long. Inner and outer rings and the 14 rollers are 440C stainless steel. The retainer, of one-piece, roller guided design, is made of silver plated, centrifugally-cast bronze. The retainer has two lugs for each roller, which are bent over the rollers for the purpose of facilitating bearing assembly. These lugs were bent away from contact with the rollers for most of the R-3 testing. This was done to prevent the lugs from rubbing the rollers and wiping off the lubricant film.

ROLLER BEARING R-4

Bearing R-4 is similar to the series R-3 bearings, with the exception of having a retainer of forged iron-silicon bronze, silver plated.

ROLLER BEARING SERIES R-5

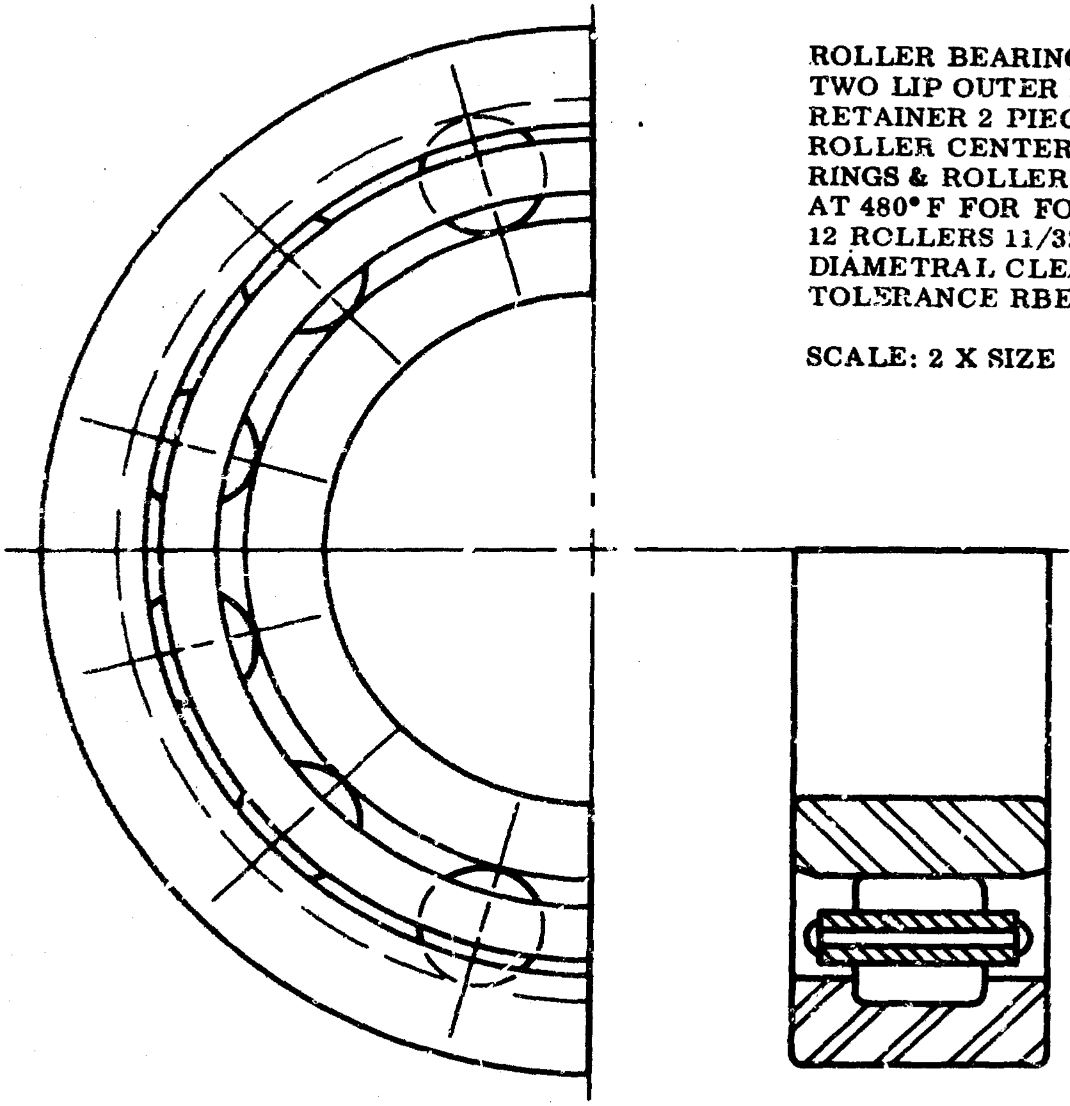
These bearings were available but testing of this design was not considered necessary because of the success obtained with the other roller-bearing designs. Bearing R-5 has a straight-through outer ring and a two-lip inner ring for guidance of the twelve rollers, which are 11/32-inch diameter and 11/32-inch long. Rings and rollers are M-50 vacuum melt tool steel. The retainer is a one piece cage, outer ring guided of S-Monel.

ROLLER BEARING SERIES R-6

Bearing R-6 is similar to R-3, but has 14 rollers, 9-mm diameter and 9-mm long, and an outer-ring guided retainer. Rings and rollers are of AISI M-50 tool steel, and the retainer is S-Monel. This bearing is shown in figure 9.

ROLLER BEARING SERIES R-7

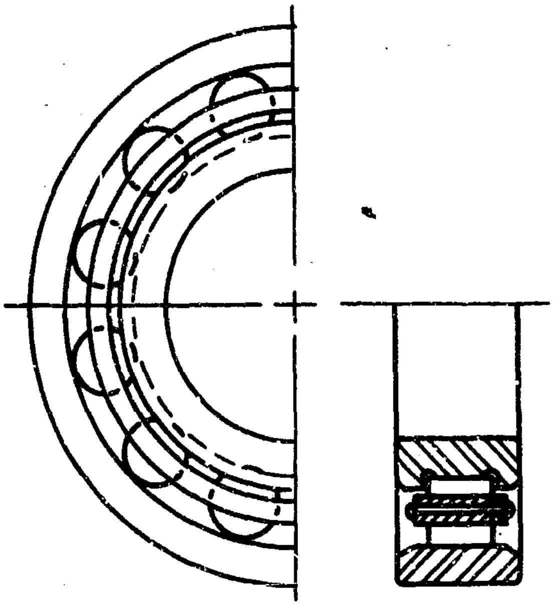
Bearing R-7 has 14 rollers, each 9-mm long and 9-mm in diameter, and a one-piece retainer of centrifugally cast bronze, silver-plated. Otherwise, the bearing is the same as the R-1 design.



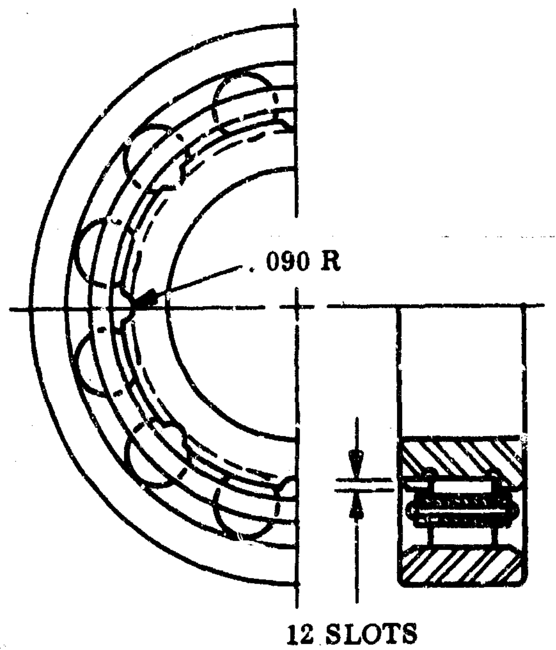
ROLLER BEARING R-1 BASIC SIZE 207
TWO LIP OUTER RING, NO LIP INNER RING.
RETAINER 2 PIECE MACHINED, RIVETED,
ROLLER CENTERED, BRASS SILVER PLATED.
RINGS & ROLLERS 52100 STEEL STABILIZED
AT 480°F FOR FOUR HOURS.
12 ROLLERS 11/32 DIA. X 11/32 LONG.
DIAMETRAL CLEARANCE .0009 TO .0013
TOLERANCE RBEC-5 GRADE

SCALE: 2 X SIZE

Figure 6. Testing Bearing R-1

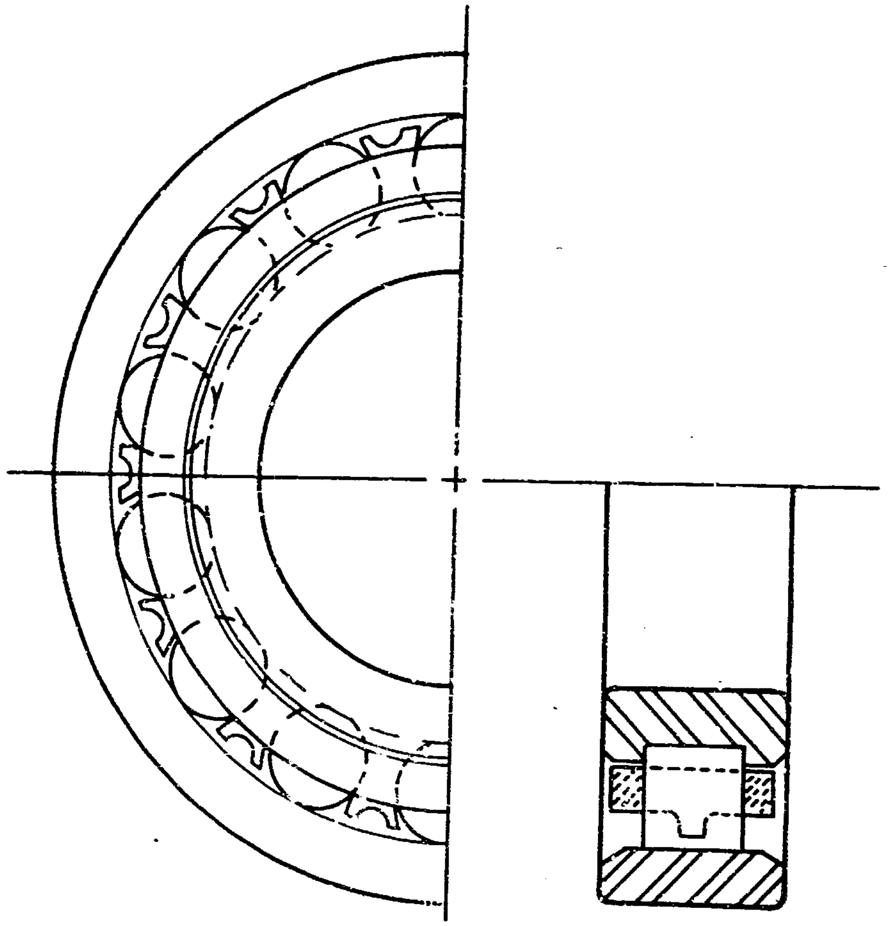


ROLLER BEARING R-2 BASIC SIZE 207
TWO LIP INNER RING, NO LIP OUTER RING
RETAINER 2 PIECE MACHINED, RIVETED,
ROLLER CENTERED, BRASS SILVER PLATED.
RINGS & ROLLERS 52100 STEEL. 12 ROLLERS
11/32 DIA. X 11/32 LONG.
DIAMETRAL CLEARANCE .001



ROLLER BEARING R-2 BASIC SIZE 207
TWO LIP INNER RING, NO LIP OUTER RING
RETAINER 2 PIECE MACHINED, RIVETED,
ROLLER CENTERED, BRASS SILVER PLATED.
RINGS & ROLLERS 52100 STEEL.
12 ROLLERS 11/32 DIA. X 11/32 LONG.
DIAMETRAL CLEARANCE .001 MODIFIED AS
SHOWN.

Figure 7. Test Bearing R-2 and R-2 Modified



ROLLER BEARING R-3, BASIC SIZE 207, TWO-LIP INNER RING, NO-LIP OUTER RING, RETAINER ONE-PIECE MACHINED, ROLLER GUIDED, CENTRIFUGALLY CAST BRONZE SILVER PLATED. RINGS AND ROLLERS 440 STAINLESS STEEL 14 ROLLERS 9 mm DIA. x 9 mm LONG, TOLERANCE GRADE RBEC-5 GRADE

Figure 8. Test Bearing R-3

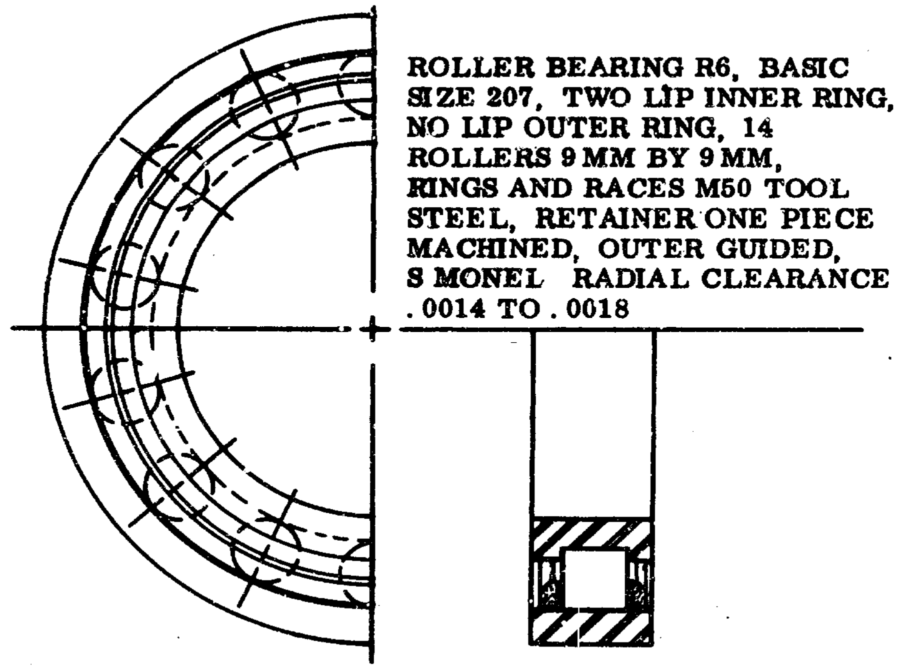


Figure 9. Test Bearing R-6

SECTION V

LUBRICANT SELECTION

Results obtained from previous development and test programs indicate that two distinct powdered lubricant mixtures are suitable for use at high temperatures. (See Reference 1 and 2.) One is a mixture of 83.33 percent graphite and 16.67 percent cadmium oxide (by weight) which is effective to 1000° F. The other is a mixture of 76 percent molybdenum disulfide (MoS_2) with 24 percent metal-free phthalocyanine (by weight) which retains its lubricating qualities up to 1200° F, but requires an inert atmosphere above 800° F to prevent oxidation of the MoS_2.

The powder lubrication system requires a method of delivering the powder to the point of lubrication. In order to keep the weight and complexity of the powder lubrication system to a minimum it was decided to utilize engine bleed air as the lubricant carrier.

The maximum temperatures that could be encountered by the lubricant were calculated to be in the range of 600-800° F. This represents the combination of bleed air temperature and the operating temperature of the bearings. In order to have the lubricant survive and function in this environment it was necessary to select a combination of powders that would successfully withstand the temperatures without any degradation of lubrication characteristics. For these reasons a mixture of graphite and cadmium oxide was selected.

SECTION VI

ENGINE TEST FACILITIES

GENERAL

The engine test stand shown in figure 10 is designed to provide qualitative and quantitative data with a minimum amount of complexity. The test stand provides means of simulating bearing loads and temperatures in the unfired engine used for testing.

In figure 10, which is a photograph of the engine on the test stand, the modified unfired J-69 engine is in the left center, with the engine exhaust (roller bearing end) at the left. The lubricator is mounted over the intake (ball bearing end) of the engine, in the top center of the photograph. The high-speed shaft with torquemeter and slip-ring assembly is visible to the right of the intake piping (covered with taped insulation), and to the right of this is the gearbox with a 5.619:1 ratio. In the lower right is the oven, which is used to heat air to the lubricator and ejectors, simulating engine bleed air temperatures. Air enters through the uninsulated piping at the right and leaves through the insulated pipes from the bottom of the oven.

ENGINE

The engine, as tested, is a basic J-69 gas turbine manufactured by the Continental Aviation and Engineering Corporation and modified by Stratos for use with graphite-cadmium oxide powder lubricant. For this series of unfired engine tests, the accessory gearbox has been removed and the combustor housing section of the turbine has been removed to enable testing with a minimum power requirement. To reduce windage losses while simulating actual weight distribution, the turbine wheel was replaced by a dummy wheel of equal weight and inertia. A steel cylinder was added to the rotor shaft to simulate the inertia of the removed compressor section. The modified rotor shaft is shown in figure 11, with thermocouple wires from the bearing inner faces to the slip-ring assembly visible on the right. The rotor assembly is balanced to within 0.04 ounce-inches in front and rear bearing planes.

Original lubricant passages and associated parts (slingers, seals) have been re-worked or completely redesigned for use of air-powder suspension lubricant flow. A cross-section of the modified engine rotor and bearing housing is shown in figure 12. Provision was made for heaters to be installed at the bearing housings to simulate thermal conditions in a fired engine. A pneumatic piston was also incorporated at the rear bearing housing to apply thrust load to the rotor shaft, simulating the thrust load encountered in the actual engine. The thrust piston and heater for the rear bearing are shown in figure 13.

POWDER LUBRICATION AND PLUMBING

The lubricator used for metering and dispensing the lubricant is described in Section VII.

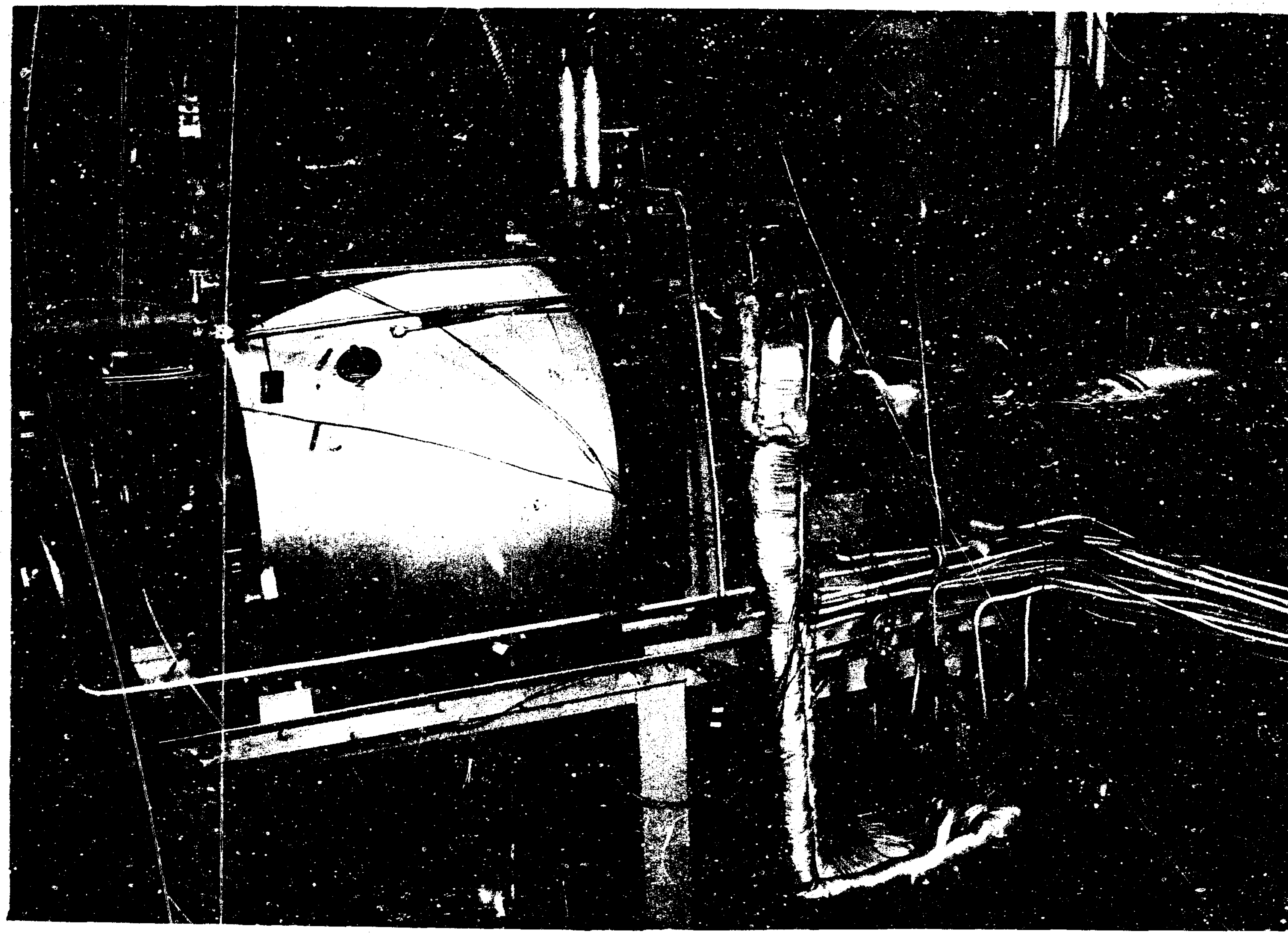


Figure 10. J-69 Engine Test Stand

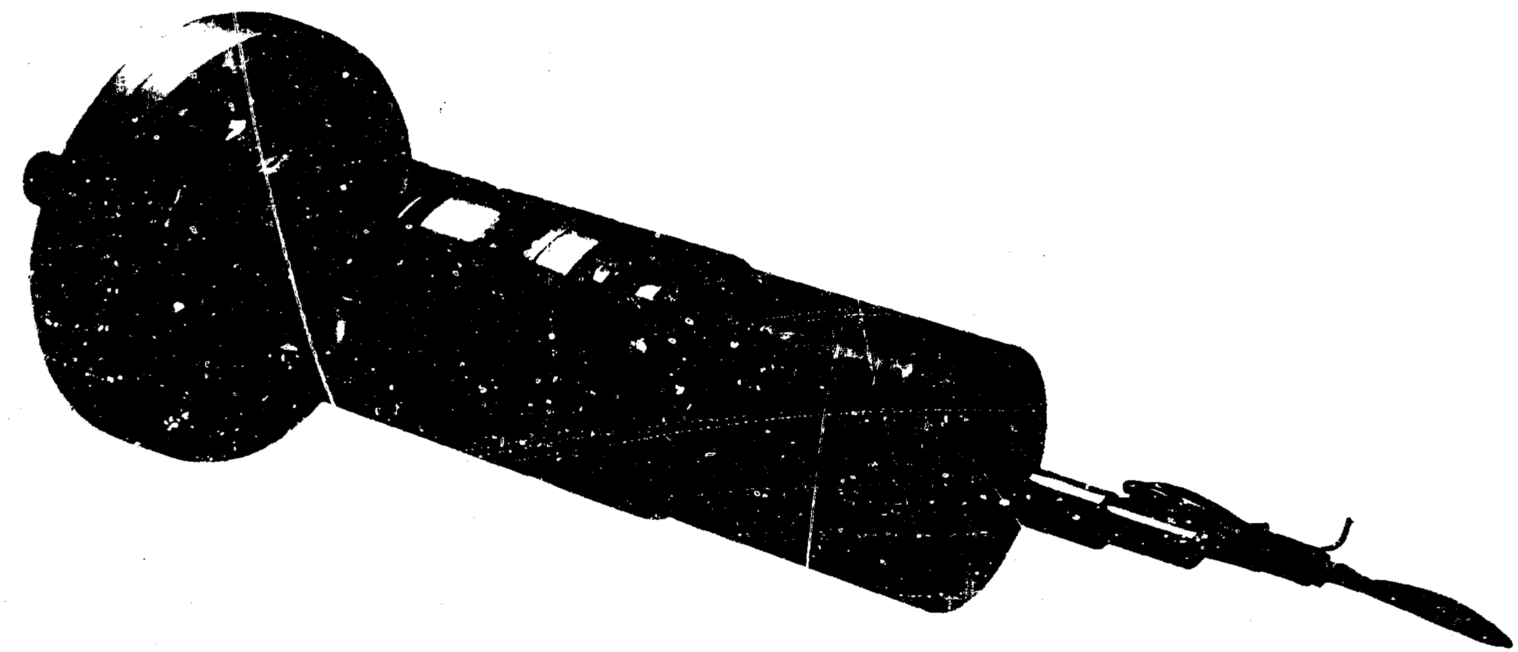


Figure 11. Rotor Shaft Modified for J-69 Engine

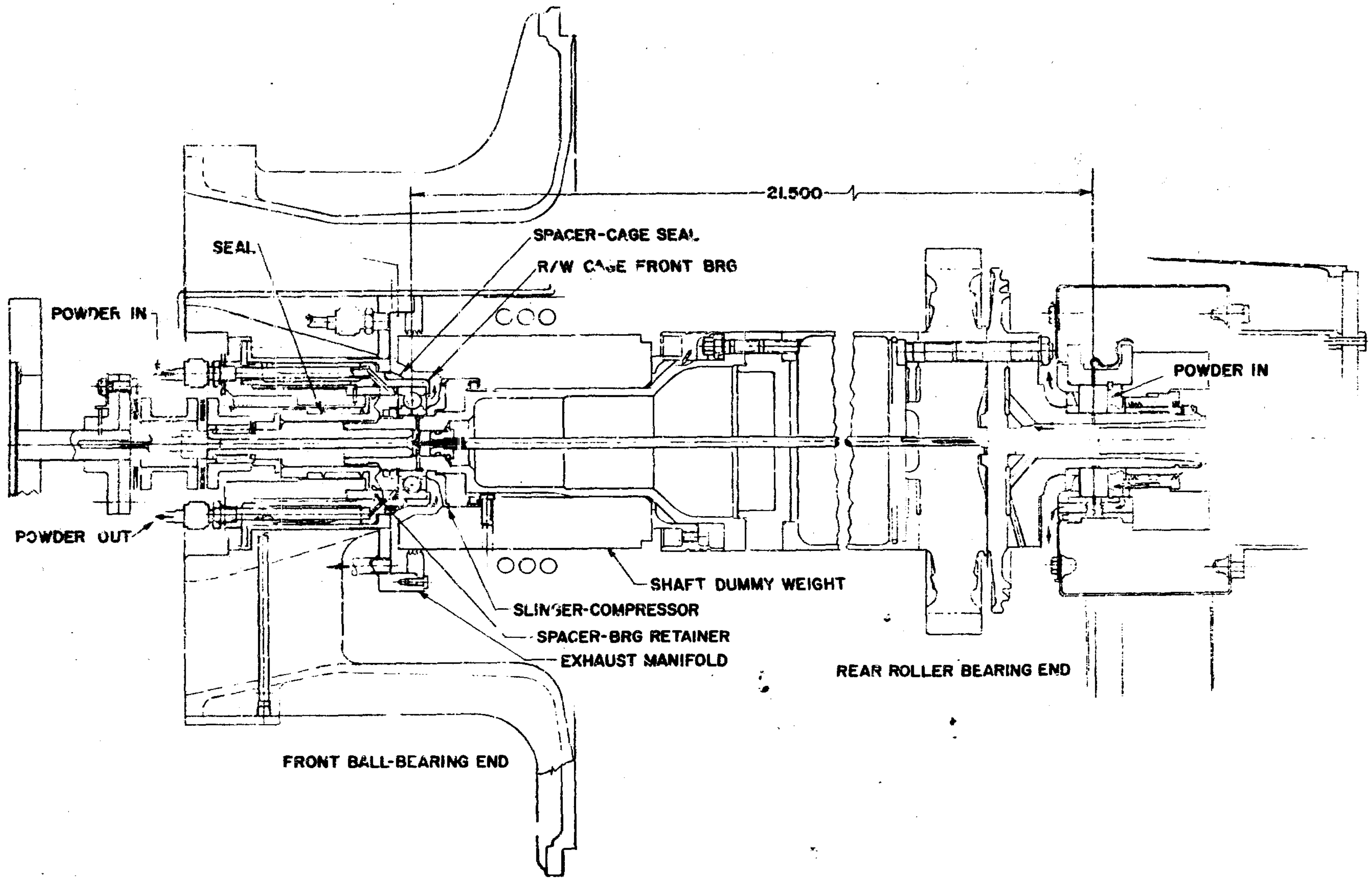


Figure 12. Engine Rotor Assembly Modified for Powder Lubrication

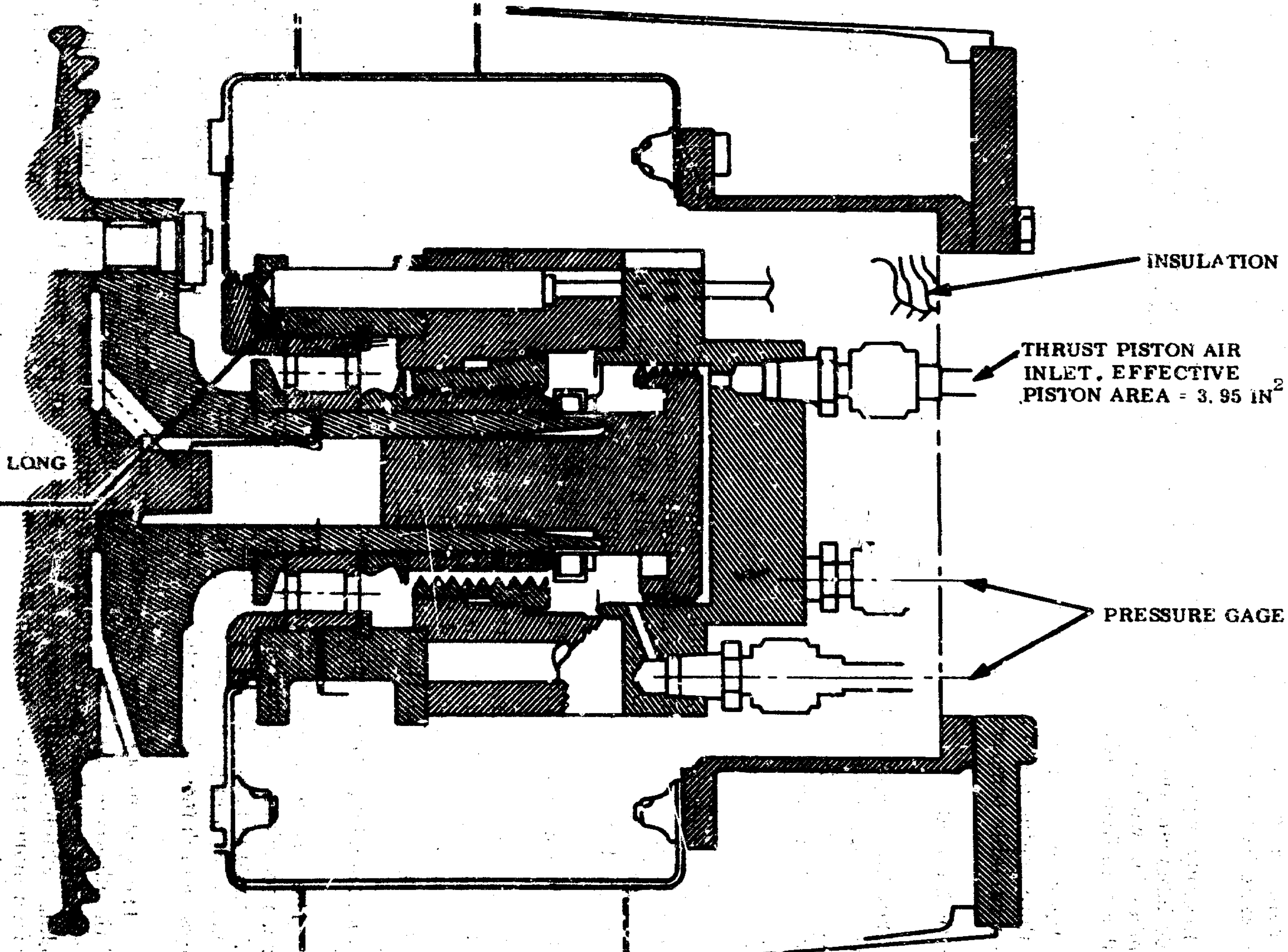


Figure 13. Rear Bearing Housing, Thrust Piston, Heaters and Seals.

Before each test run, powder is weighed in a plastic bag and then poured into the lubricator reservoir. At the end of the run, the remaining powder is poured back into the plastic bag and reweighed to determine the amount of powder used during the run.

In early engine testing a plenum chamber was installed after the lubricator to divide lubricant flow to each bearing. Later testing was made without the plenum chamber but with separate lubricator feed wheels for each bearing lubricant supply as shown in figure 14.

Provision is made for heating the air to the lubricator and each ejector to simulate engine bleed air temperatures by putting the air lines from each flowmeter into a variable temperature oven.

DRIVE MOTOR

For high speed testing to 20,000 rpm, and for testing with a thrust load on the front bearing, a 15-hp, ac motor, with variable speed gearbox, was installed to drive the J-69 engine with the use of a belt drive system. This motor and the belt drive system are shown in figures 15 and 16.

TEST RIG GEARBOX

The test rig gearbox is used to increase drive motor speed from 4500 rpm to 24,000 rpm. The gearbox is an auxiliary power turbine gearbox modified by the removal of the turbine wheel and plugging pad openings. Lubrication for the gearbox is provided by a self-contained oil sump.

TORQUE METERING SYSTEM

The torque metering system has to monitor small incremental torque variations of the test shaft assembly at fairly high speeds. Instantaneous torque readings determine the effectiveness of the lubrication system. With an unusual rise in drive motor input current, an immediate torque reading verification will enable a shutdown before impending seizure or failure. This is important, particularly with the engine rotor assembly, which weighs 86 pounds and rotates at 22,000 rpm.

The torque readout is a visual display meter with a retransmitting potentiometer for a recorder. Calibration of the system is by torque lever and dead weight and is performed periodically. The standard torque pick-up unit has been modified for the inclusion of a slip-ring assembly. Space limitations restricted this assembly to 4 rings, which provide transmission for reading 2 bearing inner-ring thermocouples.

The system operates at speeds to 20,000 rpm and has functioned reliably during all the shakedown, preliminary, and bearing stabilization tests. Due to the high speed, the brushes require frequent cleaning. The unit also has a magnetic speed pickup, which visually displays shaft speed on an rpm counter. This counter has been proven more reliable than the tach-generator on the drive motor, which is used for comparative readings. Calibration has been by strobe light.

Torque pick-up unit specifications are as follows:

- Speed, 0 to 24,000 rpm
- Torque capacity, 0 to 50 inch-pounds

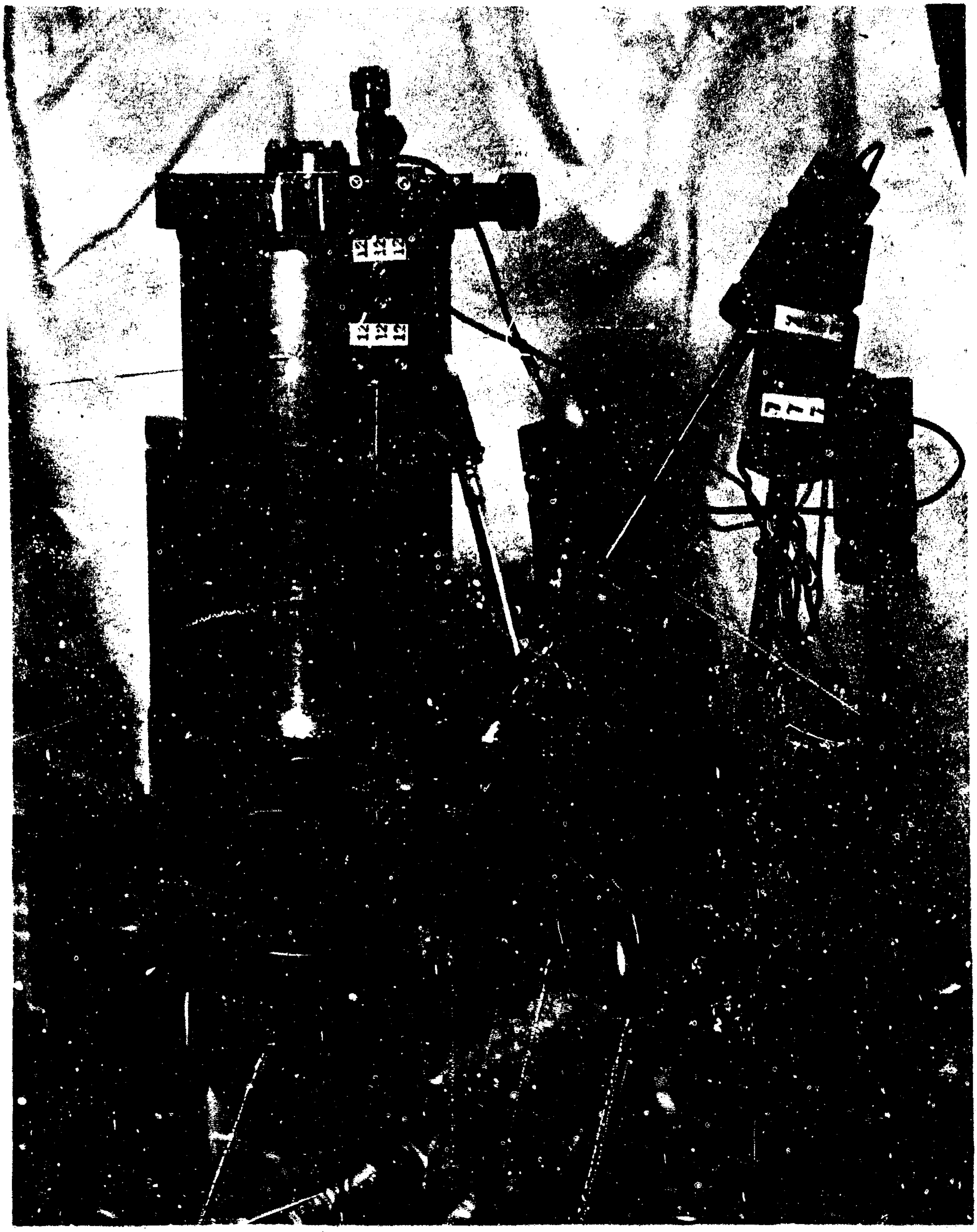


Figure 14. Lubricator Mounted on Engine

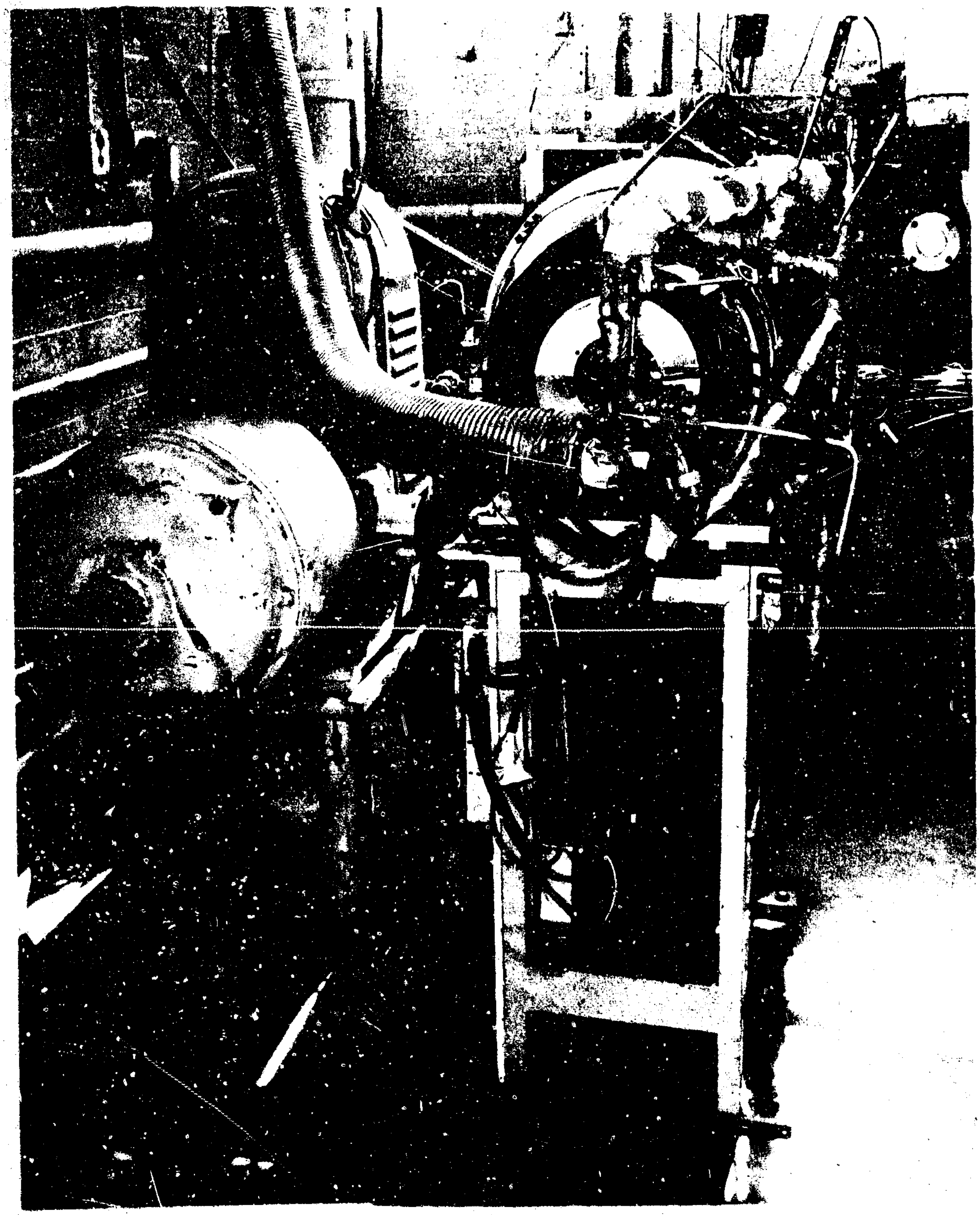


Figure 15. J-69 Test Stand with 15 Horsepower Varidrive Installation

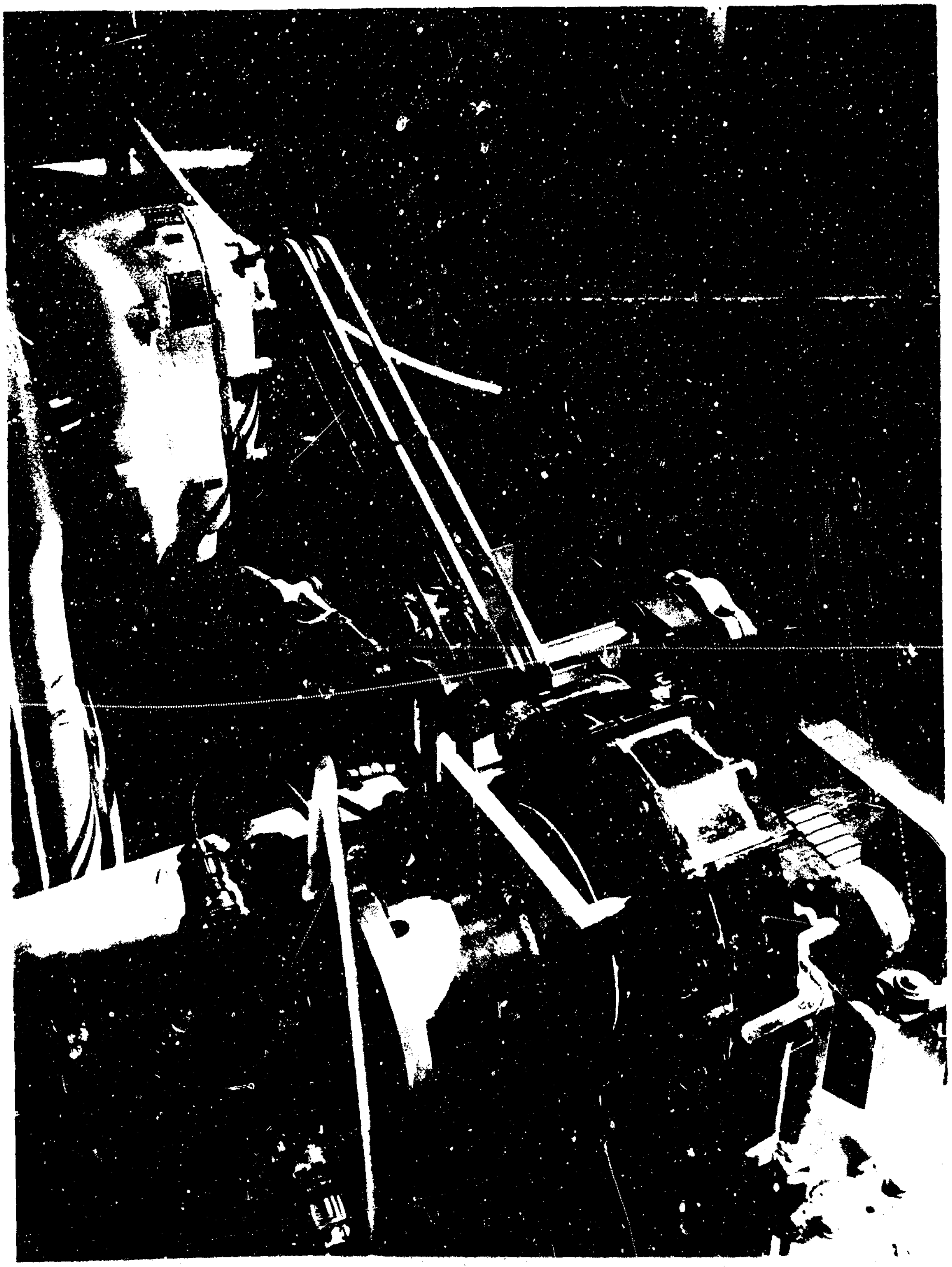


Figure 16. Test Stand Belt Drive System

- Torque sensitivity, 0.05 inch-pounds
- Linearity, 0.1 percent full scale
- Extra slip rings for thermocouples, 4
- Brush lifters, air operated

PLUMBING AND INSTRUMENTATION

Major components and all instrumentation of the engine test stand are shown schematically in figures 17 and 18. Figure 17 shows the lubrication plumbing and associated pressure gages, and figure 18 shows all electrical components, including motors, instrumentation, and thermocouples. Additional instrumentation not shown are two accelerometers, one in each engine bearing plane, used to monitor vibrations as protection against possible resonance forces induced by the modified rotor shaft and bearing mountings.

Speed control of both the main drive motor and the lubricator feed wheel is accomplished with variable-speed controlled motors.

Each separate air inlet to the lubrication system has its own flowmeter and pressure regulator. Most of the lines are 3/8-inch OD tubing and have flared AN fittings at all joints. The plumbing from the oven outlets to the respective bearing lubricant inlets is insulated with glass wool. Air temperatures and pressures are monitored at each different stage of the plumbing.

The control and instrument panel is shown in figure 19. At the upper left of the left panel board are the controls for the two variable-speed motors with associated tachometers and ammeter (for the drive motor). The square dial in the center of this panel is the torque meter readout. At the bottom of the panel are flowmeters and regulators for the air inlets (lubricator and ejectors) and above these the pressure gages for the inlet (high-pressure) plumbing.

At the lower right is the temperature readout, which is capable of switching through 20 different thermocouples. On top of this is a digital readout counter for direct readout of engine speed in rpm.

A sonic analyzer for use with the accelerometers had not been installed when this photograph was taken.

REAR BEARING HEATERS

Heaters are installed at the rear housing to enable heating of this bearing to simulate fired engine conditions. The original configuration consisted of six 600-watt capacity heaters spaced radially around the bearing housing, and one 1100-watt capacity heater inside the rotor shaft axially coincident with the rear bearing. Design changes to permit installation of the thrust piston assembly forced replacement of the 1100-watt capacity heater with one of 120-watt capacity. When the rear bearing housing was again redesigned the 120-watt heater was eliminated as it only represented 3 percent of the heating power. The heaters are connected to the power source through rheostats, which are adjusted simultaneously to permit operation of the heater group at any value to a total of 3720 watts, or 3600 watts when the 120-watt heater was eliminated.

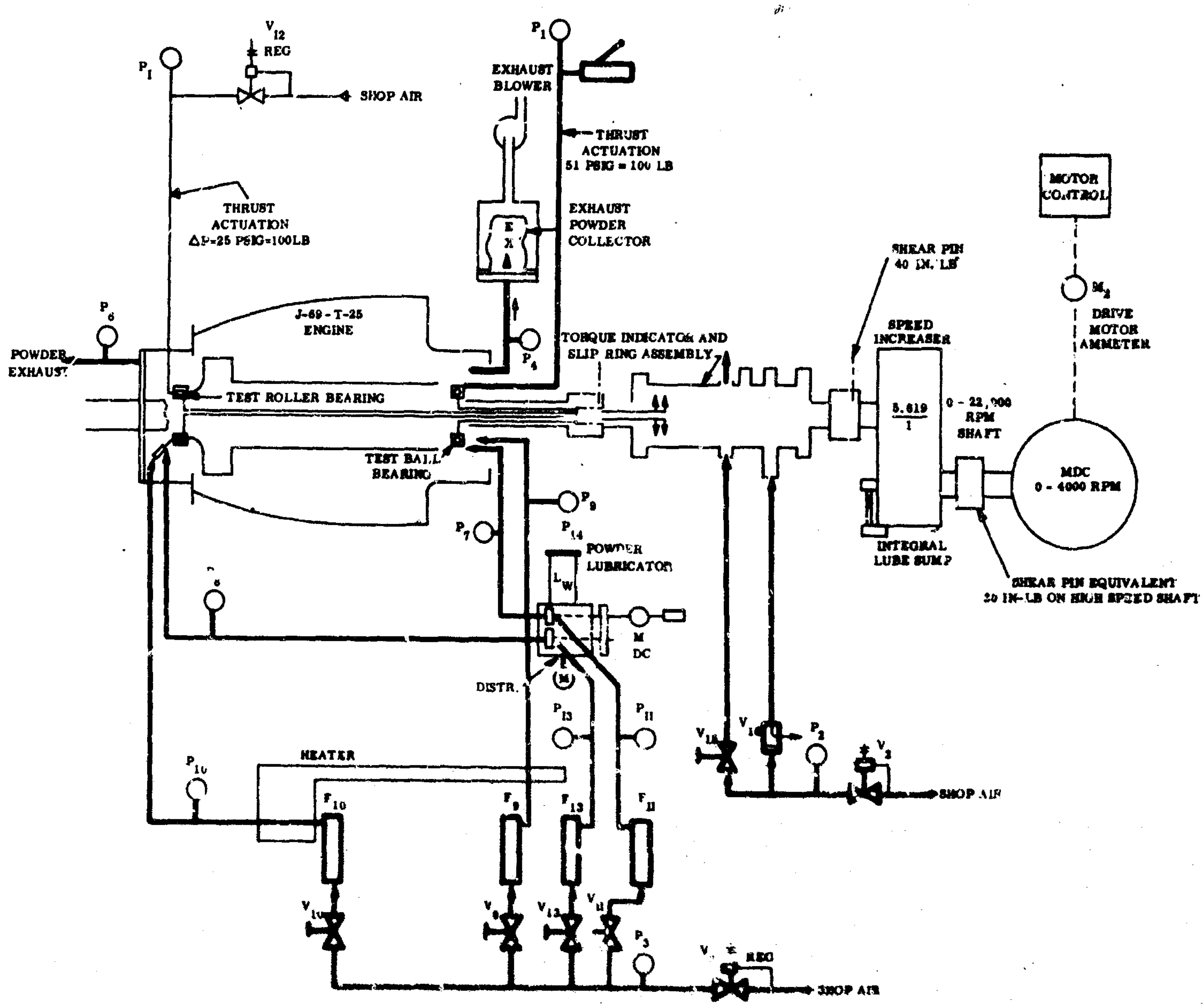
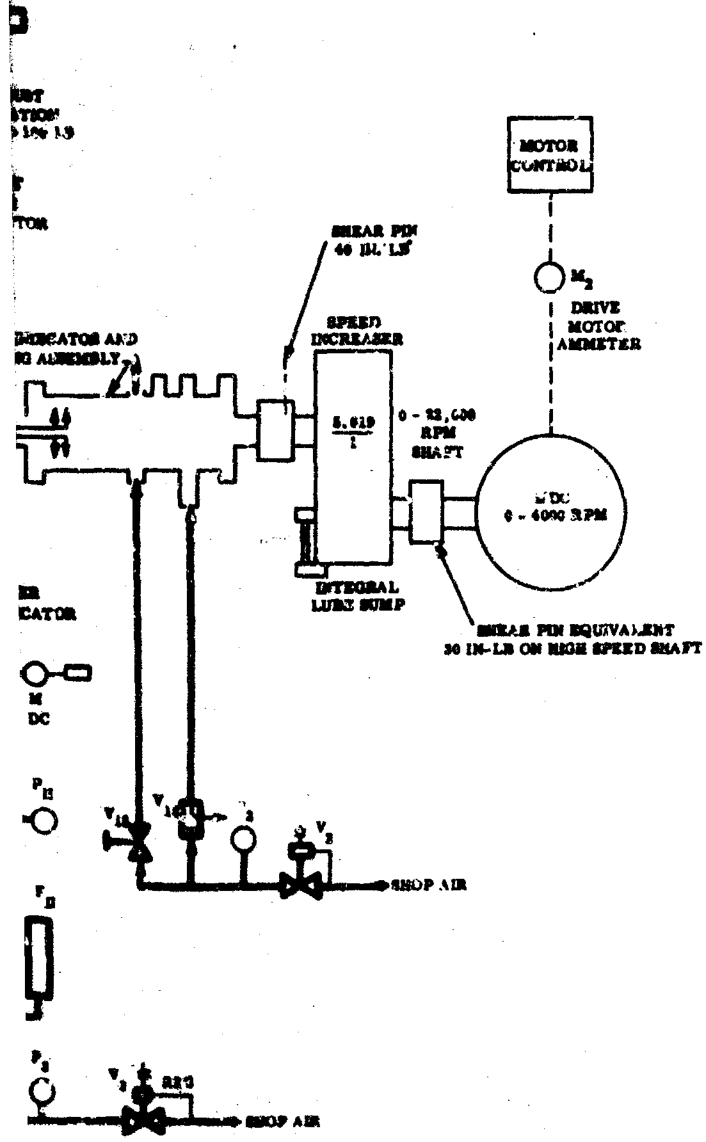


Figure 17. Schematic D
of Unbladed

A.



Note
FUNCTIONAL INSTRUMENTATION TO BE CALIBRATED AT CONSTANT FREQUENCIES PER STANDARD PRACTICE

DESCRIPTION	SYMBOL	VALUE
AIR THRUST PRESSURE	P_1	0-100 PSIG
SLIP-RING PRESSURE	P_2	0-100 PSIG
FLOWMETER INPUT AIR PRESSURE	P_3	0-60 PSIG
BALL BEARING EXHAUST PRESSURE	P_4	± 5 PSIG
ROLLER BEARING EXHAUST PRESSURE	P_6	± 5 PSIG
BALL BEARING CARRIER AIR PRESSURE	P_7	± 10 PSIG
ROLLER BEARING CARRIER AIR PRESSURE	P_8	± 10 PSIG
BALL BEARING EJECTOR AIR PRESSURE	P_9	0-40 PSIG
ROLLER BEARING EJECTOR AIR PRESSURE	P_{10}	0-40 PSIG
LUBRICATOR INPUT AIR PRESSURE/BALL BEARING	P_{11}	0-40 PSIG
LUBRICATOR INPUT AIR PRESSURE (ROLLER BEARING)	P_{13}	± 10 PSIG
BALL BEARING EJECTOR AIRFLOW	F_9	0.3 LB/MIN
ROLLER BEARING EJECTOR AIRFLOW	F_{10}	0.3 LB/MIN
LUBRICATOR AIRFLOW (BALL BEARING)	F_{11}	0.3 LB/MIN
LUBRICATOR AIRFLOW (ROLLER BEARING)	F_{13}	0.3 LB/MIN
SLIP-RING AIR MASTER VALVE	V_2	
BEARING LUBE AIR MASTER VALVE	V[illegible]	
BALL BEARING EJECTOR AIR VALVE	V_9	
ROLLER BEARING EJECTOR AIR VALVE	V_{10}	
LUBRICATOR AIR VALVE (BALL BEARING)	V_{11}	
THRUST ACTUATOR AIR MASTER VALVE	V_{12}	
LUBRICATOR AIR VALVE (ROLLER BEARING)	V_{13}	
SLIP-RING COOLING AIR VALVE	V_{15}	
SLIP-RING BRUSH LIFTER RELIEF VALVE	V_{16}	

Figure 17. Schematic Diagram of Air System for Test Rig of Unbladed J-69 Engine

B.

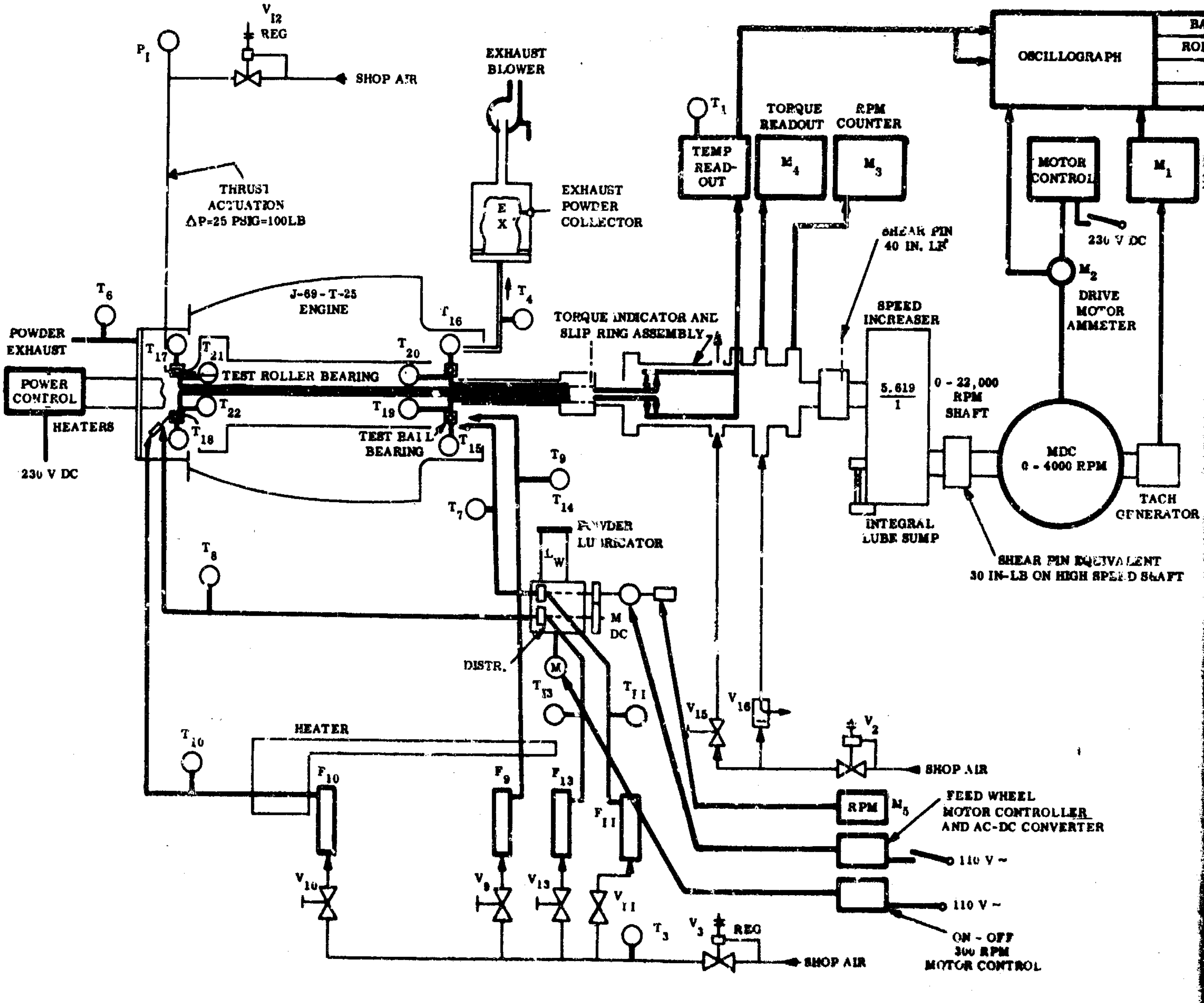
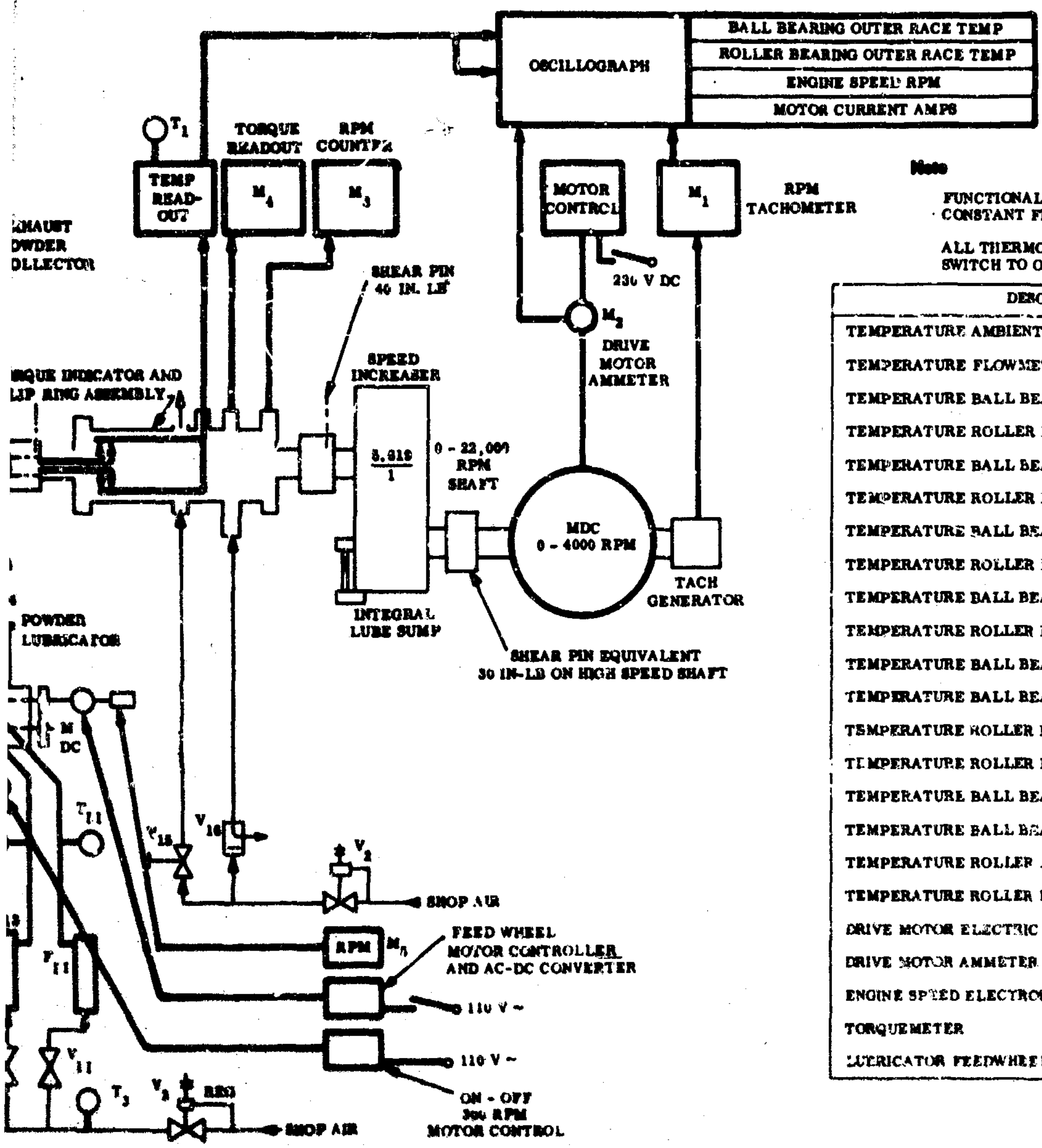


Figure 18. Schematic Dia
of U..bladed J

A.



Note

FUNCTIONAL INSTRUMENTATION TO BE CALIBRATED AT CONSTANT FREQUENCIES PER STANDARD PRACTICE

ALL THERMOCOUPLES FEED THROUGH SELECTOR SWITCH TO ONE TEMPERATURE READOUT INSTRUMENT

DESCRIPTION	SYMBOL
TEMPERATURE AMBIENT	T_1
TEMPERATURE FLOWMETER INPUT AIR	T_3
TEMPERATURE BALL BEARING EXHAUST AIR	T_4
TEMPERATURE ROLLER BEARING EXHAUST AIR	T_6
TEMPERATURE BALL BEARING CARRIER AIR	T_7
TEMPERATURE ROLLER BEARING CARRIER AIR	T_8
TEMPERATURE BALL BEARING EJECTOR AIR	T_9
TEMPERATURE ROLLER BEARING EJECTOR AIR	T_{10}
TEMPERATURE BALL BEARING CARRIER INPUT AIR	T_{11}
TEMPERATURE ROLLER BEARING CARRIER INPUT AIR	T_{13}
TEMPERATURE BALL BEARING OUTER RACE ACTIVE	T_{15}
TEMPERATURE BALL BEARING OUTER RACE SPARE	T_{16}
TEMPERATURE ROLLER BEARING OUTER RACE ACTIVE	T_{17}
TEMPERATURE ROLLER BEARING OUTER RACE SPARE	T_{18}
TEMPERATURE BALL BEARING INNER RACE ACTIVE	T_{19}
TEMPERATURE BALL BEARING INNER RACE SPARE	T_{20}
TEMPERATURE ROLLER BEARING INNER RACE ACTIVE	T_{21}
TEMPERATURE ROLLER BEARING INNER RACE SPARE	T_{22}
DRIVE MOTOR ELECTRIC TACH	M_1
DRIVE MOTOR AMMETER	M_2
ENGINE SPEED ELECTRONIC COUNTER	M_3
TORQUEMETER	M_4
LUBRICATOR FEEDWHEEL TACH	M_5

Figure 18. Schematic Diagram of Electrical System for Test Rig of Unbladed J-69 Engine

B.

Figure 19. Instrumentation Control Panel

SECTION VII

POWDER DISPENSING SYSTEM

GENERAL

Concurrent with bearing rig and engine testing, studies were performed to determine the best means of metering and dispensing the powder lubricant. The results of these studies are described in the following paragraphs.

MANIFOLD PRESSURE

It was found that at the air mass flow rate of 0.03 pound per minute used for all bearing rig testing, the pressure in the plumbing between lubricator and bearing housing was negligible. With a vacuum cleaner on the bearing exhaust port, the pressure in the plumbing was 5 in. Hg vacuum. This was evidently sufficient to keep the powder from settling out of the airstream and collecting on the inside of the pipe.

FEED WHEEL AIRFLOW

Under normal lubricator operating conditions, during bearing rig testing, 0.03 pound per minute flow at 0.25 psig, a slight positive pressure had been noticed in the powder reservoir of the lubricator, indicating air leakage at the feed wheel. In order to study this leakage better, lubricator testing was conducted with an air inlet pressure of 25 psig. At this pressure, leakage was noticed around the circumference and face of the feed wheel. Various wheels were tried with different feed wheel clearances in the wheel housing, but this did not alleviate the leakage problems. An air bleed line was run from the lubricator inlet to the powder reservoir in an attempt to equalize pressure around the feed wheel. No consistent results were obtained with this method and it was discontinued. The position of the air jet hole to the feed wheel was changed from the original design (figure 20) so that the airflow through the lubricator would never be completely blocked by the feed wheel (figure 21). This appeared to alleviate the air leakage around the feed wheel.

POWDER DENSITY

Measurements were taken to determine if the packing density of powder in the feed-wheel bucket had a significant effect on weight flow. The bucket was hand packed loosely with powder and the powder used was weighed. The procedure was repeated with the powder tightly packed. The powder weight was approximately the same for both tests, showing that as long as the bucket is filled completely, the powder weight in the bucket is constant.

WEIGHING METHODS

Because of the small quantity of lubricant used for each test run (in the order of several grams maximum), the method used to determine the weight of powder used during a run is very critical. Because of the relative weight of the lubricator housing compared to the powder used (9 pounds vs several grams, respectively) it was found that accurate flow determinations could not be made by weighing the

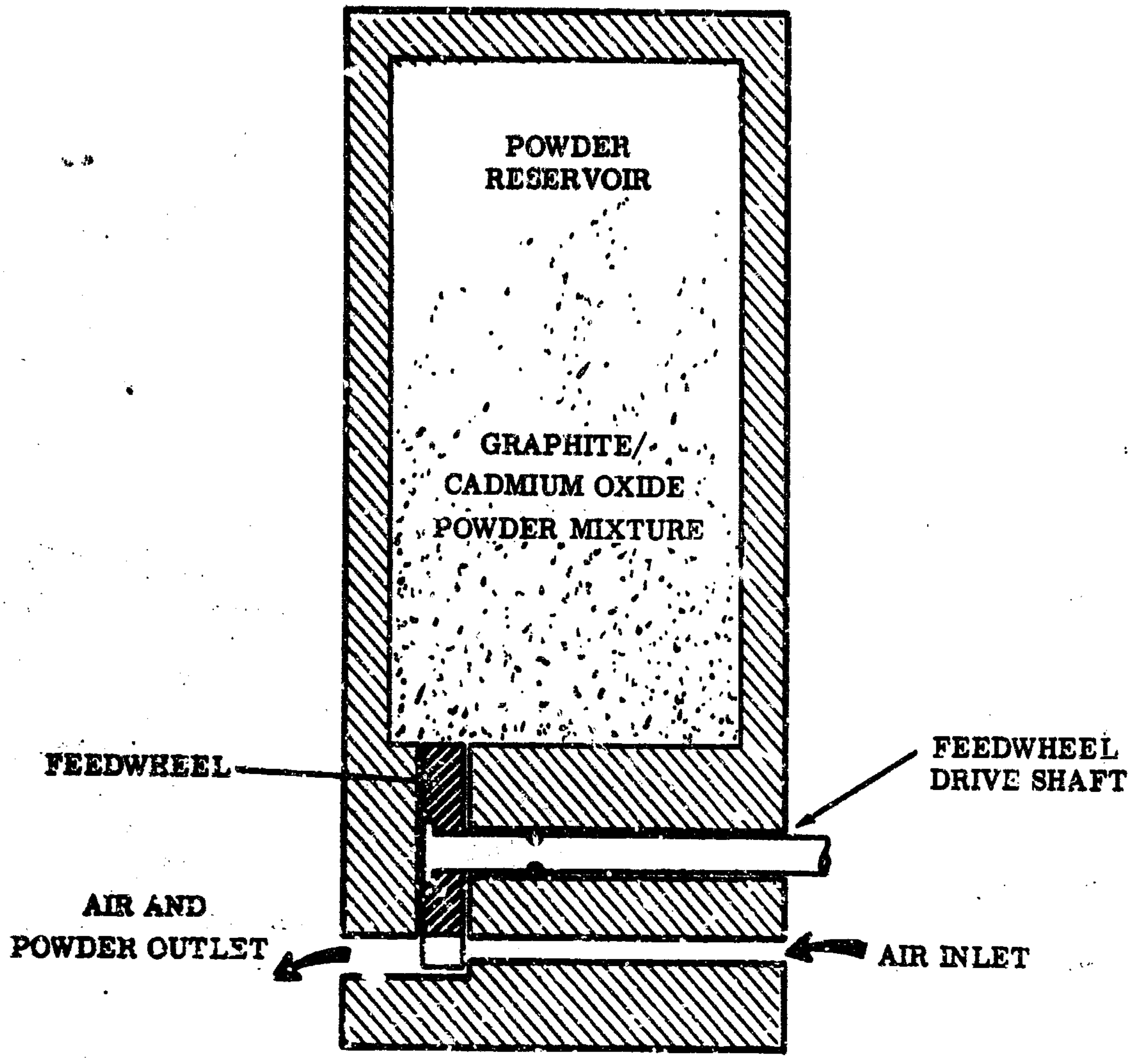


Figure 20. Lubrication Schematic Cross Section

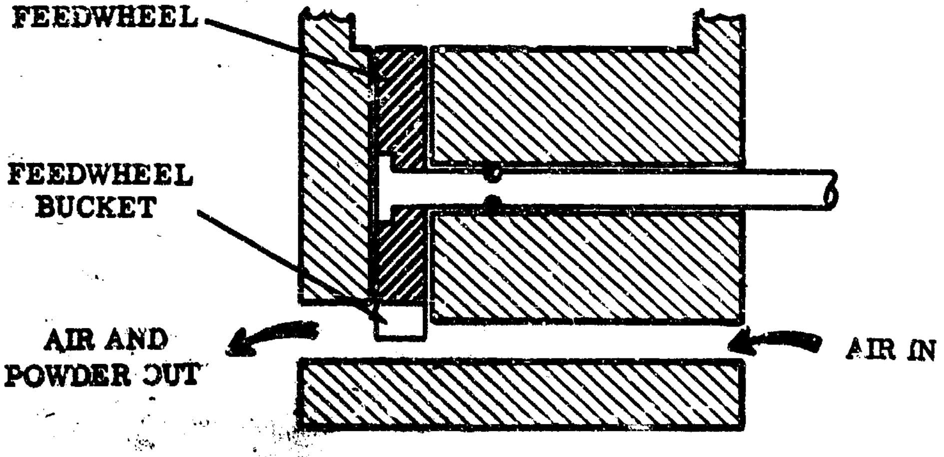


Figure 21. Feed Wheel Air Inlet Modification

powder while in the lubricator. The best weighing method was found to be as follows:

Before a run, the powder is weighed in a plastic bag and then poured into the lubricator. After the run, the remaining powder is poured from the lubricator into the bag and again weighed. The difference in weights before and after the test run is the powder used in that run. The power collected in the exhaust powder collector was not weighed. It was demonstrated that in the development of the above method of weighing the powder that an additional check was not necessary.

LUBRICATOR DESIGN

The test rig lubricator used for metering and dispensing the lubricant is shown in figure 22, and an exploded view of the lubricator in figure 23. This positive displacement type of feeder is similar to that used effectively in previous powder lubricant programs (Reference 1 and 2).

The lubricator has a cartridge type reservoir containing the powder. The powder, which is weighed separately before and after running, is metered by a bucket wheel, containing 24 buckets, located beneath the canister. Each bucket of the feed wheel is filled with powder as it passes the canister and is emptied after 180 degrees of wheel travel by a carrier gas jet. Feed rate is adjustable by varying the speed of the feed wheel or changing the bucket configuration of the wheel, which is readily accessible. The agitator is installed inside the powder reservoir and is run at constant speed to prevent powder vortexing and buildup in the reservoir.

As alternates to the design described in the preceding paragraph, two new designs for a lubricator were developed and preliminary tests were performed on them. One design consisted of a syringe mechanism filled with powder in which a piston forced the powder through an orifice into the carrier airstream. The major difficulty with this design involves the accurate pumping of a given amount of powder per unit time throughout the operating time of the test.

The other design consisted of compressed powder being carried out of a reservoir into the airstream by means of a screw thread (as used in some coal conveyors). The major problem encountered with this design is packing of powder in the reservoir to the extent that it can not be forced into the screw conveyor at a fixed rate. As successful results were obtained with the original feed wheel design, further development of the alternate designs was not considered necessary.

LUBRICANT DISTRIBUTION

In the bearing test rig, only a single bearing required powder lubricant, and this was readily supplied by one carrier line from the single feed wheel lubricator. For engine testing, however, two separate bearings required powder lubricant, and a means had to be provided for such lubricant distribution.

Earlier testing (Reference 4) had shown that a simple "Y" pipe was not an effective powder flow divider, because pressure differentials between the two outlet lines from the "Y" caused an extreme variation in relative powder flow between the two lines. A distributor was devised utilizing a swirl-type plenum chamber with ejectors on the plenum outlet lines. Preliminary tests of the plenum chamber indicated that it was relatively insensitive to differences in pressure between the two outlet lines,

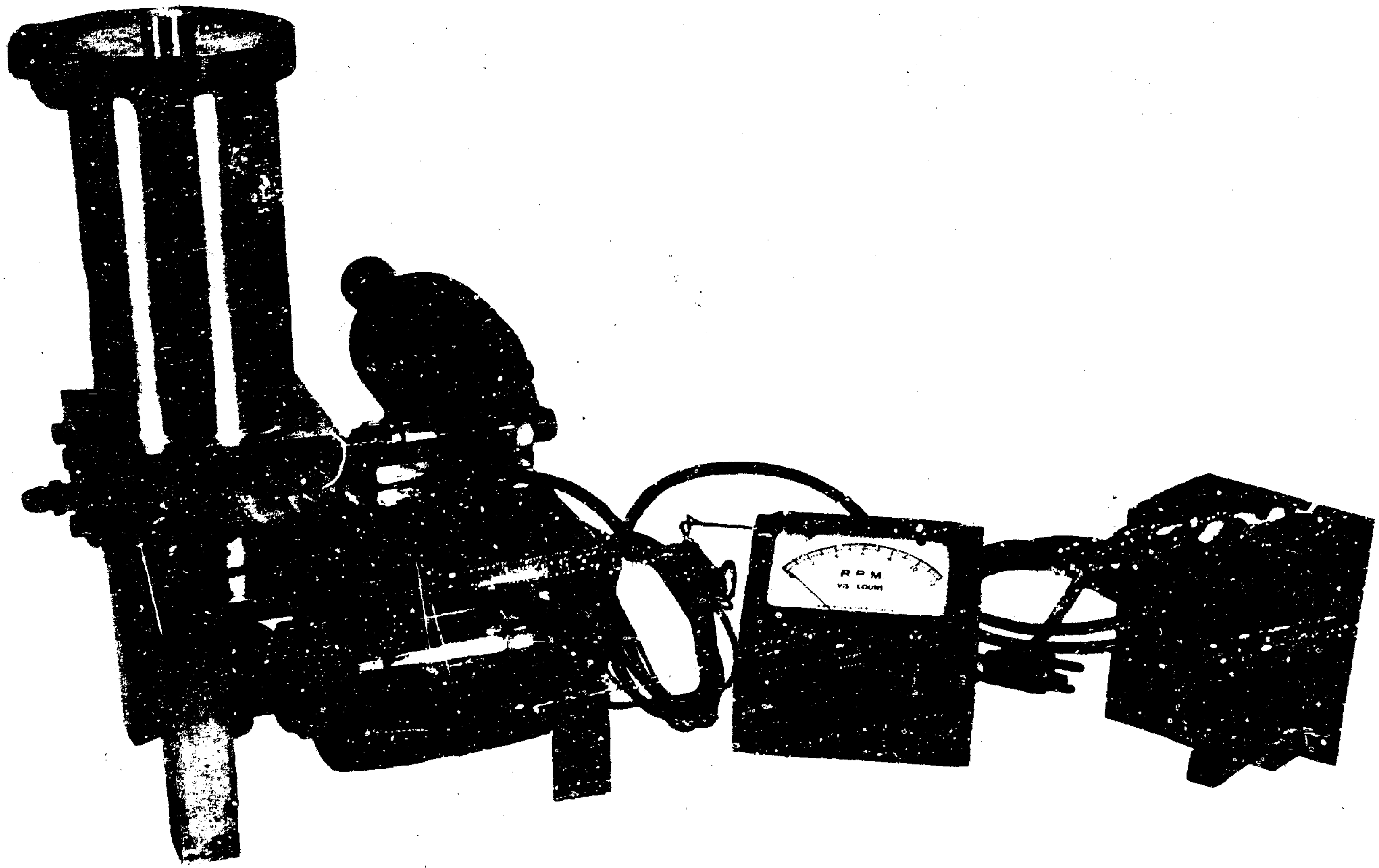


Figure 22. Test Rig Lubricator

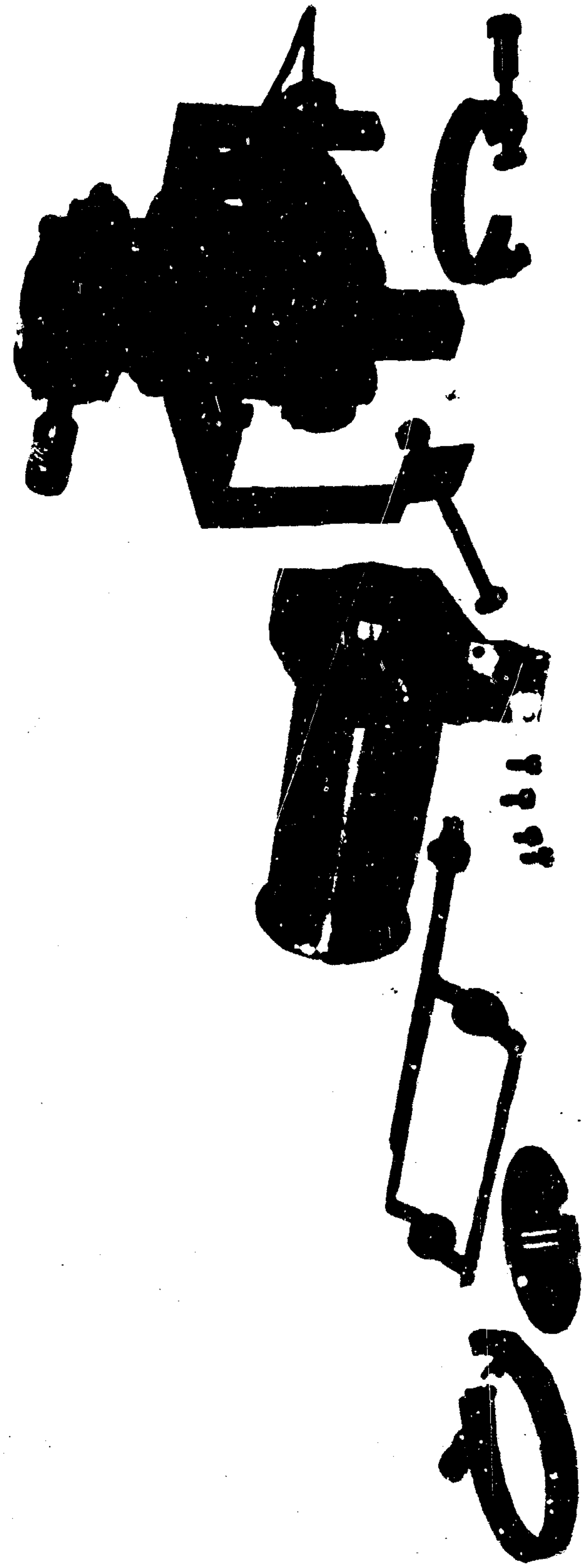
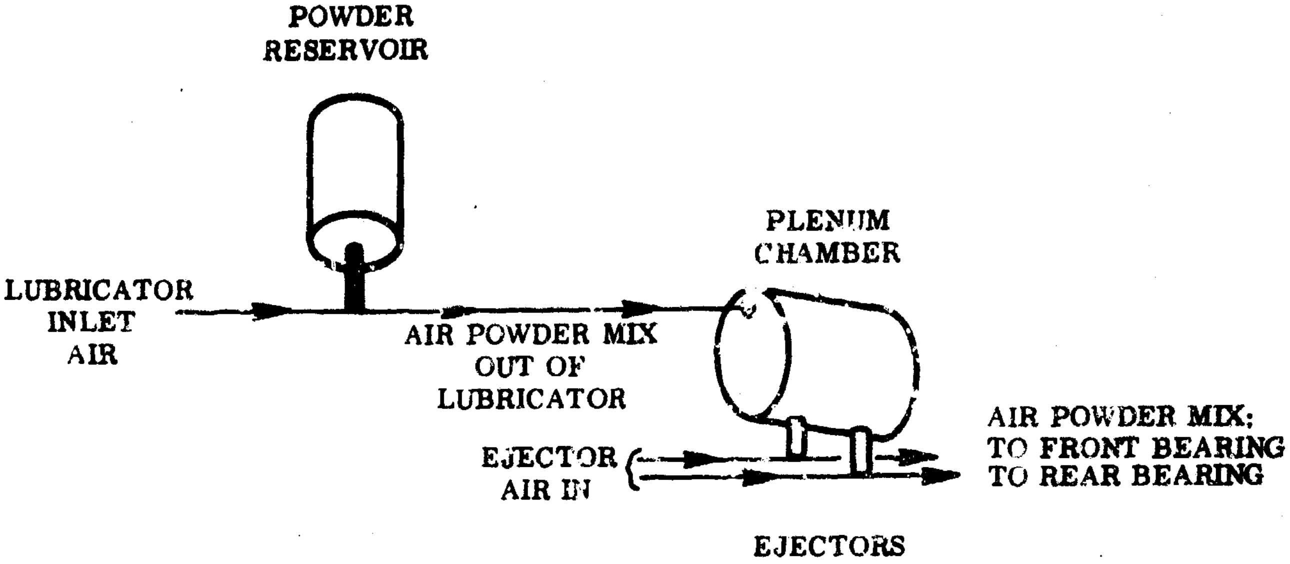


Figure 23. Exploded View of Lubricator

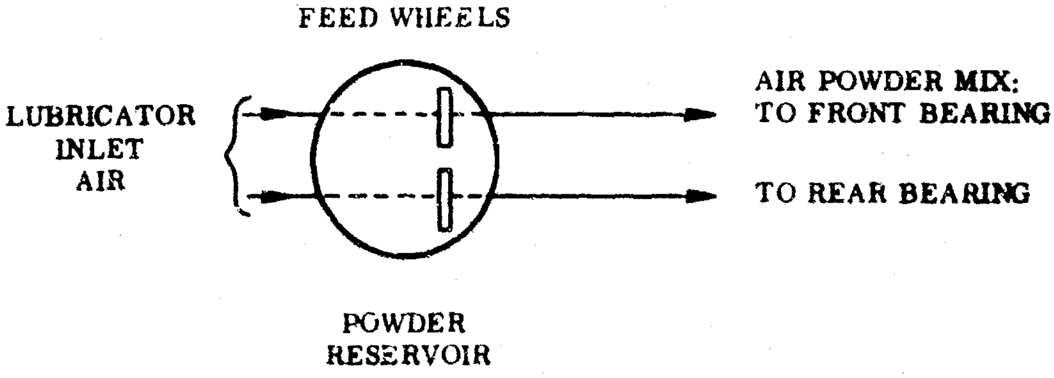
therefore this distribution system was used for the first few engine tests. Further testing, however, revealed that, even with a plenum chamber accurate division of powder flow could not be obtained. This was due to differences in downstream pressures acting through the plenum and affecting each other, and to powder falling out of suspension in the plenum chamber and building up in the lower half of the chamber.

A successful solution to adequate lubrication of both bearings was achieved by providing one feed wheel and air supply line for each bearing. The original lubricator was modified by placing a second feedwheel next to, but separate from, the original feed wheel, and gearing both wheels together to operate at the same speed. With proper sizing of the feed wheels, this method was found successful for providing equal powder flows to both bearings.

The comparison between the unmodified lubricator (single feed wheel lubricator with plenum chamber) and the modification incorporating two separate feed wheels is shown in figure 24. The modified lubricator is shown mounted on the J-69 engine in figure 14.



a) Original Design



b) Modified Design

Figure 24. Modified Lubrication Distribution Method

SECTION VIII

ENGINE TEST PROCEDURE

GENERAL

The J-69 engine main rotor runs on two bearings, a ball bearing at the intake end for both radial support and thrust load, and a roller bearing at the exhaust (turbine) end for radial support. The front bearing operates in an ambinet temperature environment, and the roller bearing in a high-temperature environment created by the high-temperature exhaust gases. Engine performance characteristics have been described in detail in Section III of this report and appendix I and II of reference 4.

TEST OBJECTIVES

The final objective of this engine test program is the operation of an unfired engine with powder lubricated bearings at conditions of speed, load, and temperature expected in a fired engine with powder lubrication. Operation at these conditions was achieved in a series of test runs, the general sequence of which is as follows:

- Testing at low speed (8000 rpm) with progressively higher environmental temperatures at the rear bearing housing. No thrust load on front bearing. (Maximum fired engine rear bearing running temperature, arrived at by adding a contingency of 15 percent of the difference between ambient temperature and operating temperature as calculated in the heat transfer analysis, is estimated to be about 530° F. This is the maximum value that was used for high-temperature testing.)

- Testing at progressively higher speeds, from 8000 to 20,000 rpm, with rear bearing at estimated fired engine heating conditions. No thrust load on front bearing.

- Testing at low speed with increasing thrust load on front bearing. Rear bearing at ambient environmental temperature.

- Testing at progressively higher speeds, from 8,000 to 20,000 rpm with the thrust load on the front bearing.

TEST DATA

During each test run, the following data were recorded:

- Shaft speed

- J-69 engine torque

- Carrier-[illegible] temperature, both bearings

- Carrier [illegible] pressure, both bearings

- Carrier-air flow, both bearings
- Outer-race temperature, both bearings
- Ambient room temperature
- Lubricator feed wheel speed
- Rear-bearing heater voltage
- Thrust load on front bearing

In addition, air pressures, temperatures, and flow rates were measured at various critical parts of the plumbing (into and out of lubricator, at ejectors, etc.).

Cycling frequency of recording data was 10 minutes for critical measurements, such as bearing temperatures and engine torque, and 60 minutes for general data, such as ambient temperature and air-flow rates, which remain constant throughout the duration of a test run.

TEST DURATION

Each test was run until bearing temperatures stabilized (indicated by no more than 4° F change in one hour) or until temperature, torque, or noise indicated rough running with the possibility of bearing failure. Testing was stopped after 4 hours running time if the bearing had not achieved 1 hour of stabilized running by that time. Room ambient temperature was between 70 and 85° F.

SECTION IX

SEAL DESIGN AND EVALUATION

CONVENTIONAL ENGINE SEALS

In the conventional oil lubricated engine, the bearing-cavity shaft seals are used to minimize the leakage of pressurized lubricating oil.

At the front bearing a face-type seal is used to prevent lubricant leakage into the main air intake to the compressor. This low-leakage seal is used for the following reasons. Although the cavity is vented to ambient through the accessory case, the entering oil is pressurized at 20 to 40 psi above ambient. The spent oil is not scavenged from this location and is drained into the accessory gearbox.

At the rear bearing, the leakage requirement is less rigid and labyrinth seals are used. A scavenge pump removes the oil from this position, which is also vented to ambient.

POWDER LUBRICANT FRONT BEARING SEALS

An investigation of front bearing seals was made on two types of face seals, mechanical labyrinth and labyrinth seals. The labyrinth seal was selected for this application.

LABYRINTH SEAL

The labyrinth is a simple sealing device. It restricts leakage by a controlled gap to reduce flow area and it forms a series of orifices, each equal to the gap or clearance area. The labyrinth clearance must allow for the bearing radial clearance. At the front bearing, this clearance is nominally 0.0025 inch at room temperature. This requirement, coupled with a short axial space limitation, has made it necessary to study means to control the leakage flow through the labyrinth.

Two labyrinth seals are shown in figure 25. Study number one is a stepped labyrinth. The touch points are stepped diametrically to minimize carryover past these points. However, with the short axial space allowance, it was necessary to improve this method, as shown in figure 26 and seal study number 4. With this method a positive suction head, induced by the exhaust ejector, will drain the seal of powder.

Study number two, figure 25, utilizes buffer air to change leakage paths. The principal advantage of this method is that cooling air to the bearing cavity will help remove powder buildup at the seal side of the bearing. A separate air jet from the annular chamber directed to the bottom portion of the outer ring prevents excessive powder buildup. These two types were successfully tested on the engine test rig. These seals are not subjected to elevated thermal stresses, and do not have centrifugal and frictional rubbing loads. The shaft and component assembly is simplified by the removal of axial travel limits and preload required with a face-type seal.

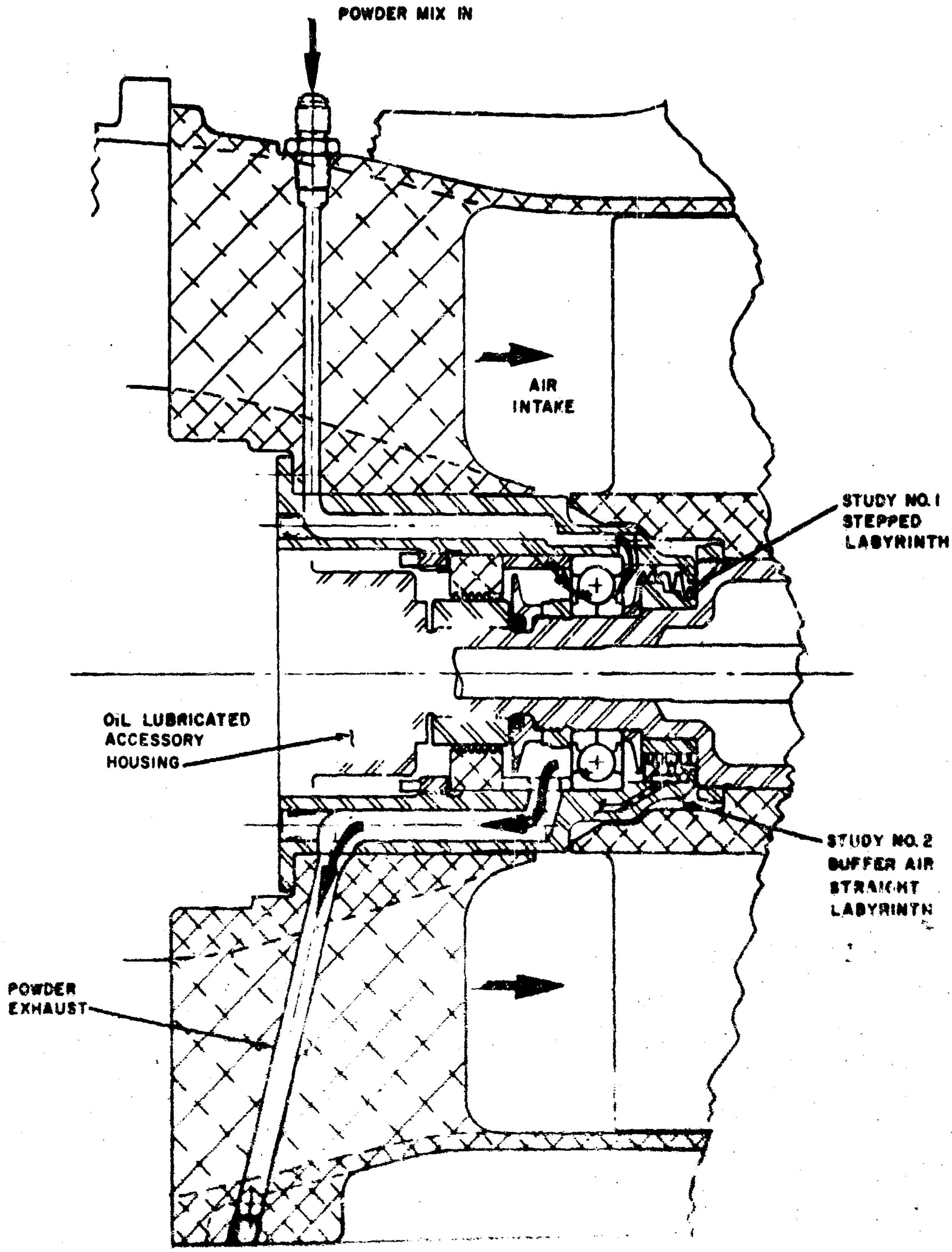


Figure 25. Labyrinth Type Seals

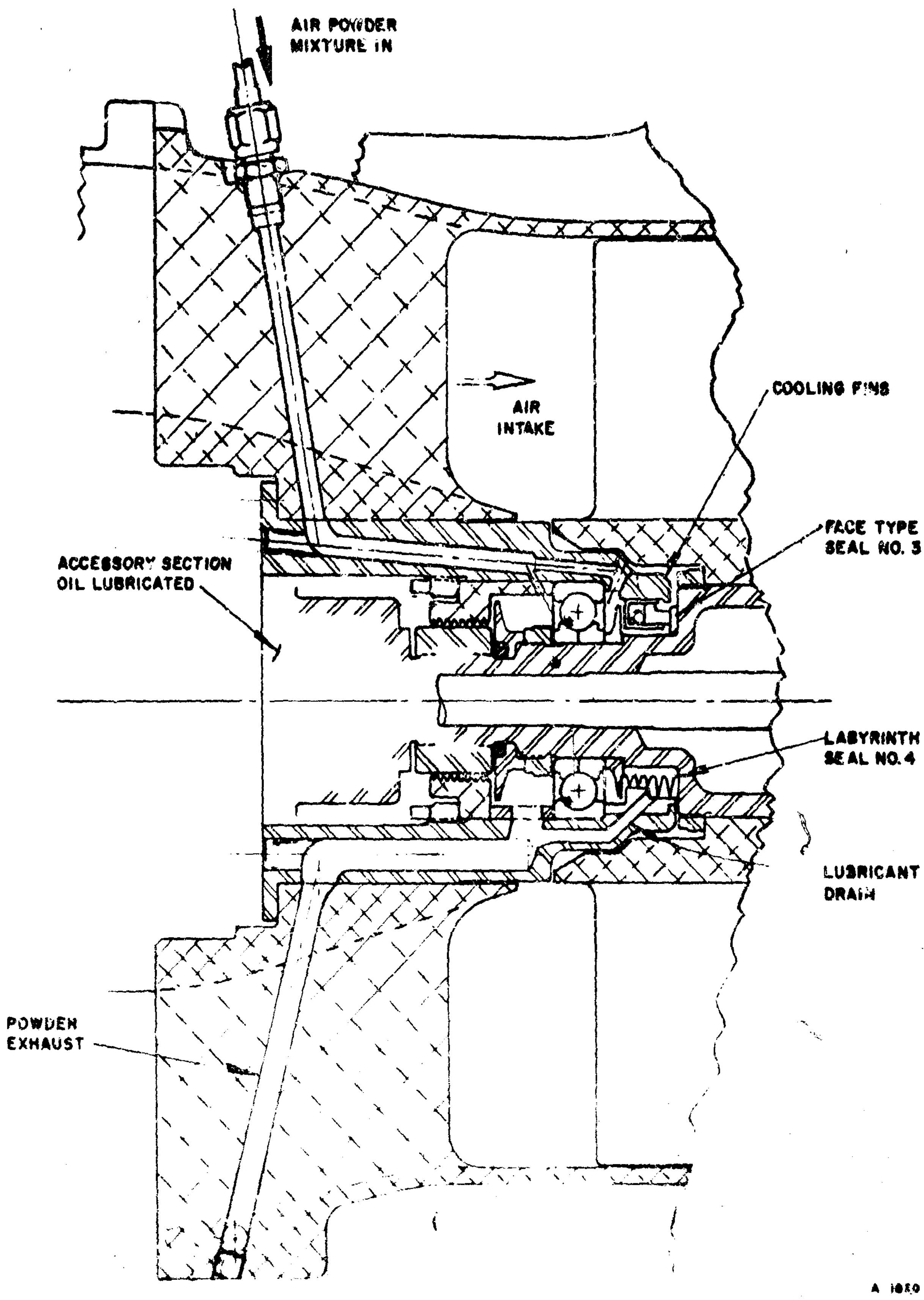


Figure 26. Alternate Seals

The material for the initial seals were aluminum for the grooved body and steel for the co-mating sleeve.

CARTRIDGE TYPE FACE-TYPE SEAL (FRONT BEARING)

Face-type seals are useful where low leakage limits are required. These seals can be run at fairly high temperatures and high rubbing speeds, but dry-gas seals require large and rigid support housings. The face-type seal has the following disadvantages:

1. Installation time.
   On complex assemblies, such as a turbojet engine, shims and special methods and tooling are required to position stator and rotor parts with the proper initial preload.

2. Heat generation.
   At this bearing location, a comparatively thin rotor (0.218 inch) abuts the inner race of the bearing. With a restricted flow of cooling air, the rotor surface speed, 11,000 fpm, will produce undesirable frictional rubbing heat, which will be transmitted into the inner ring of the bearing.

3. Requirement for squareness at operating conditions.

4. Vibration and axial movement.
   The secondary seal (high-temperature "O" ring) has to seal and provide free axial movement at elevated temperatures.

5. Gradients.
   The rotor and stator rings must withstand deflections due to pressure and mechanical loading. These deflections will produce frictional gradients.

METALLIC BELLOWS FACE SEAL

Investigation of a metallic bellows seal was made. This type of seal has a rotationally locked secondary seal, reducing friction at the primary seal. The primary seal rotates at approximately half shaft speed. These are two highly desirable characteristics.

The disadvantages of this type seal are as follows:

1. High cost.

2. Critical installation dimensions.

3. Gradients.
   The bellows will operate at high stress levels due to pressure, and mechanical and thermal range. These gradients may induce axial and radial instability.

4. Without viscous dampening surrounding the bellows severe high-frequency oscillations can occur at the seal face.

5. The heat rejection to the bearing cannot be determined without testing or dry running this type of seal at operating conditions.

FLOATING MECHANICAL SEAL (FRONT BEARING)

No detailed investigation was made of this type of seal. The requirement for extremely close diametrical and flatness tolerances makes this a costly seal for our investigation.

POWDER LUBRICANT REAR BEARING SEALS

Labyrintn Seal (Rear Bearing)

At this seal location, with the throw-away spent powder system, the leakage requirement is eased considerably. The only requirement is that the seal will ensure penetration of the powder into the bearing.

The conventional engine rear bearing labyrinth seal will be used at this location. Installation of this seal is shown on figure 27.

The forward labyrinth step seals used on the engine at the rear bearing are not required and may or may not be used, depending on test buildup requirements.

UNFIRED ENGINE SEAL DESIGN

Front Bearing

Figure 27 shows the seal design as installed in the unfired test engine. A combination of a rotating slinger and labyrinth seal is located at the front of the bearing and a rotating slinger is located at the rear. The slingers create a positive head of air and also tend to centrifugally accelerate the powder particles to the outside of the respective cavities. A supplementary air jet directed into the front of bearing assists in maintaining a continuous flow of powder through the bearing. The rear slinger accelerates the particles into the exhaust manifold where the particles are discharged into the exhaust tube.

Rear Bearing

Figure 13 shows the installation of the rear bearing seal in the unfired test engine. The conventional engine rear bearing labyrinth seal and slinger were used without change. The forward labyrinth step seals were eliminated and a rotating slinger was substituted in its place.

FIRED ENGINE SEAL DESIGN

The seal configuration incorporated in the building of the J-69 Fired Engine is described in Section XI.

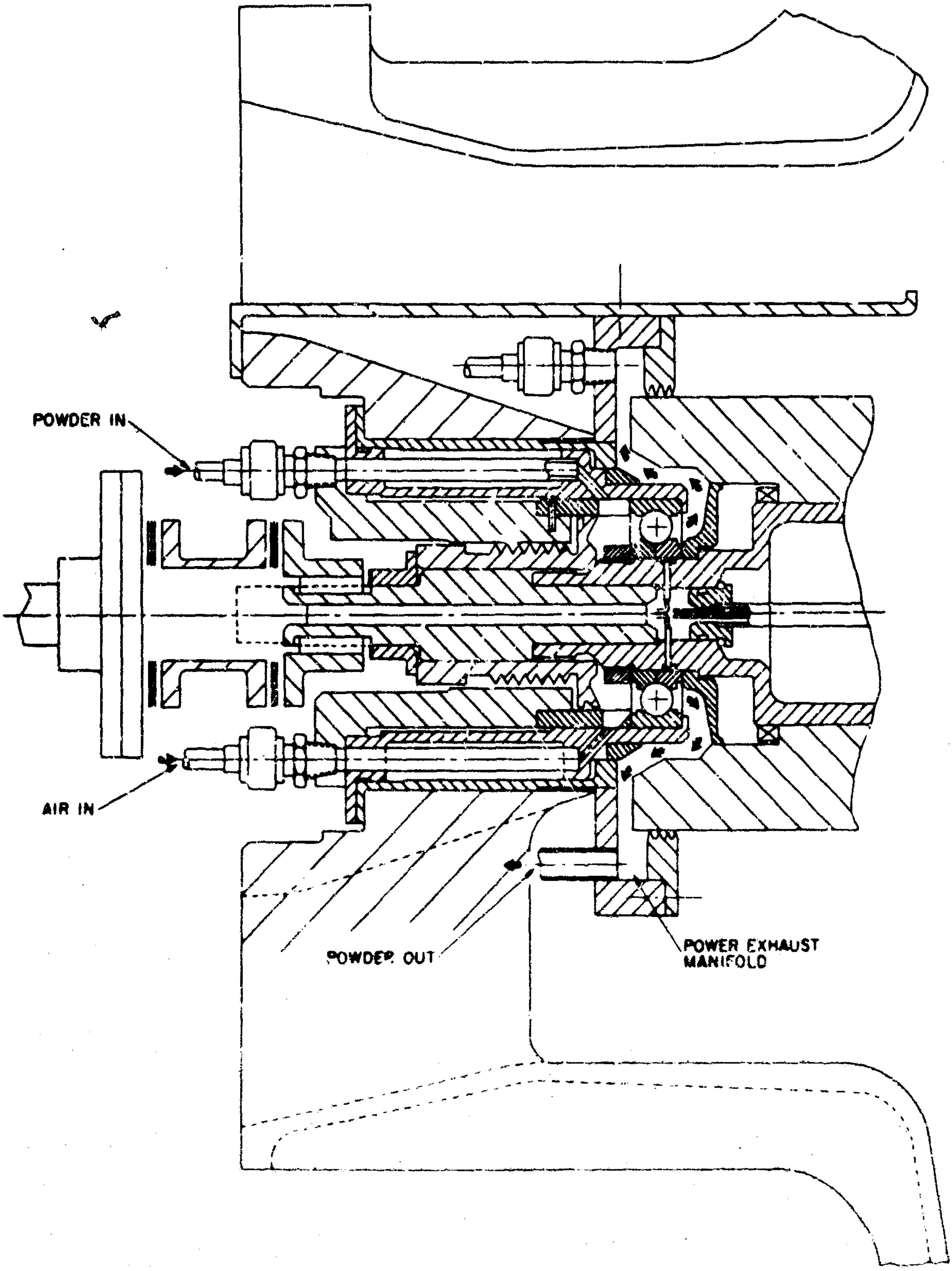


Figure 27. Front Bearing Seals

SECTION X

UNFIRED ENGINE TESTING

GENERAL

In accordance with the general test sequence outlined in section VIII, testing was successfully performed under the required conditions as follows:

- Testing at simulated rear bearing operating conditions was completed with the successful performance of a test run at 20,000 rpm with stabilized temperature of 515° F.

- Testing was successfully performed at 8000 rpm with up to 500 pounds thrust on the front bearing. The ball bearing failed at 12,000 rpm with 500 pounds thrust.

- Testing at simulated front bearing operating conditions of 50 pounds thrust load and 20,000 rpm with heated (bleed air temperature) carrier air was successfully performed with two different bearings of different designs.

- A ball bearing was successfully operated at 20,000 rpm with heated carrier air and 100 pounds thrust load.

A summary of all unfired engine testing during the period covered by this report is shown in Table III. A summary of all previously run tests is given for reference in Table IV. Detailed description of these tests may be found in Reference 4. Prior to this reporting period, a total of 38 unfired engine tests had been made at various speeds and temperature conditions. Before test E-39, a new drive system was installed to improve performance at high load and speed conditions. This drive system was described in section VI. This section will summarize all of the significant test results obtained during this report period. Detailed descriptions of the test results with specific emphasis on those tests which achieved the programs objectives are presented in the appendix.

PHASE 2 ENGINE TESTING (TEMPERATURE VERSUS SPEED)

Phase 2 engine testing consisted of running the unfired J-69 engine at varying speeds from 8,000 to 20,000 rpm with heated carrier air and heated rear bearing. Figure 28 indicates the results of the temperature versus speed tests using the bearing series B-5 and R-6. The most significant result is the shape of the roller bearing temperature curve. At 8,000 rpm the bearing temperature is the highest, drops off significantly at 12,000 rpm, and starts to climb as the speed increases. The drop in temperature from 8,000 rpm to the higher speeds is attributed to the greater air circulation around the bearing housing caused by the increased drag of the turbine rotor disc. After this series of tests were completed, an additional run at 8,000 rpm was made to verify the original test data. The results correlated very well: 540° F for the initial test and 516° F for the check test. Horsepower, torque and ball bearing temperatures increased with an increase of speed. Power required to drive the engine increased by about 575 percent as the speed was increased from 8,000 to 16,000 rpm.

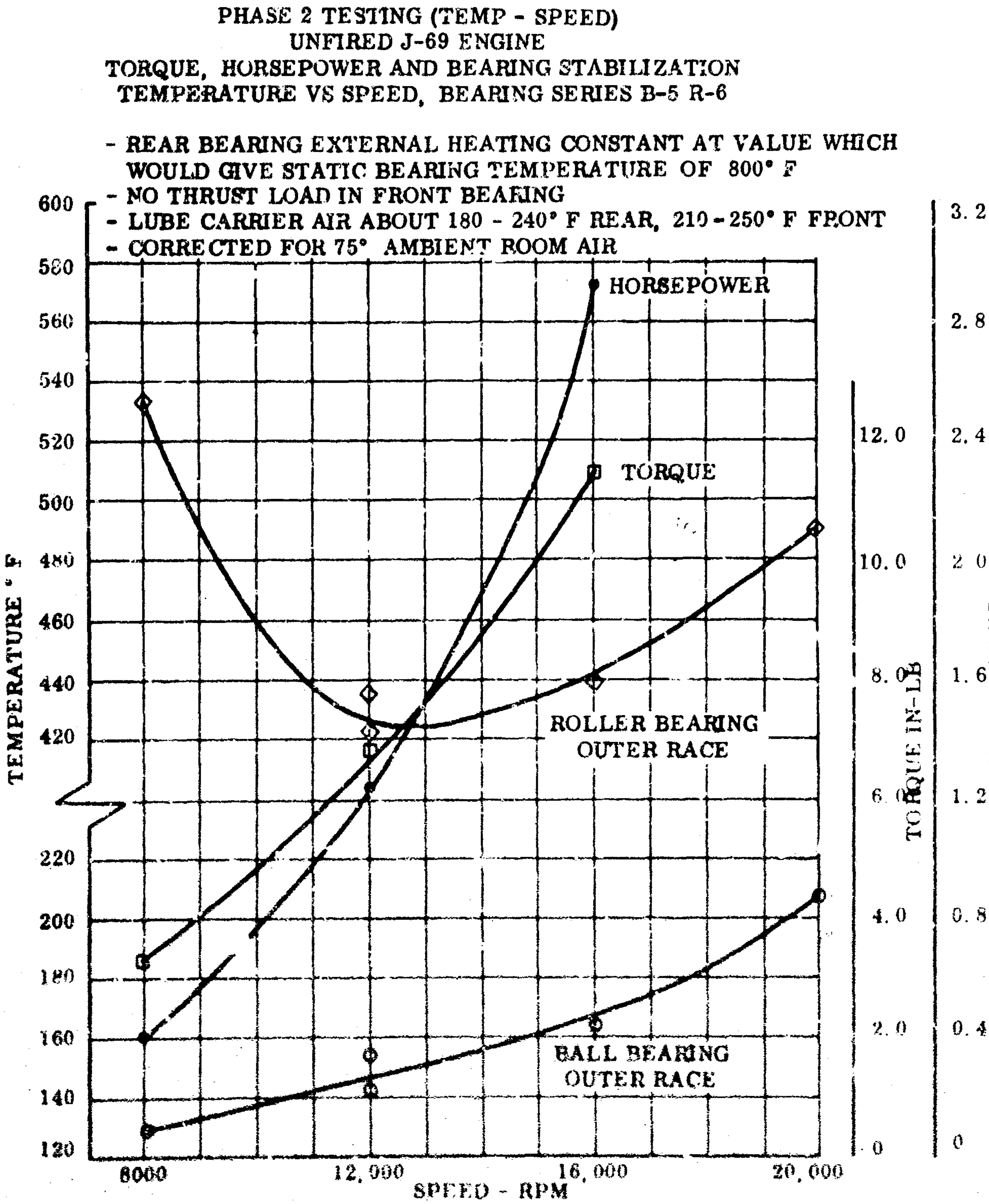


Figure 28. Summary of Phase 2 (temperature-speed) Engine Testing

TABLE III. SUMMARY OF BEARING TESTS

Test No.	Ball (Front) Bearing S/N	Roller (Rear) Bearing S/N	Speed (rpm x 10^{-3})	Avg Torque (in-lb)	Thrust Load Front Bearing (lb)	Rear Bearing Heating (watts)	Stabilized Temp (*) Ball Bearing (° F)	Stabilized Temp (*) Roller Bearing (° F)	Airflow Ball Bearing (lb/min)	Airflow Roller Bearing (lb/min)	Powder Flow Ball Bearing (gm/min)	Powder Flow Roller Bearing (gm/min)	Time of Run (hr)	Remarks
E-39	B-5a	R-6a	12	6.7	0	763	155	471	0.24	0.18	0.009	0.009	2.0	Checkout of new drive system. External heating of 763 watts.
E-40	B-5a	R-6a	16	11.5	0	648	188	466	0.24	0.18	0.010	0.010	4.0	Stabilization at 16K rpm. Started with heating at 763 watts. Heating reduced to 648 watts after 2.3 hours. Bearings ran well.
E-41	B-5a	R-6a	20		0	648	231	515	0.24	0.12	0.015	0.015	2.5	Stabilization at 20K rpm. Bearings ran well.
E-42	B-5a	R-6a	8	3.5	0	648	140	516	0.24	0.18	0.013	0.013	2.0	Calibration run with new thrust piston housing at rear bearing.
E-43	B-5a	R-6a	8	3.8 4.2	0 70	0 0	103 119	113 109	0.24 0.24	0.18 0.18	0.015 0.018	0.015 0.015	2.0 1.2	Start of thrust load testing. Thrust increased from 0 to 70 pounds after 2.0 hours. Rear bearing temperature dropped because of air leakage around the thrust loading piston.
E-44	B-5a	R-6a	8	4.3	100	0	115	90	0.24	0.18	0.018	0.018	2.0	Continued testing at increased thrust.
E-45	B-5a	R-6a	8	4.8	250	0	122	91	0.24	0.19	0.018	0.018	1.0	Continued testing at increased thrust. Bearings ran well.
E-46	B-5a	R-6a	8	4.9	423	0	151	90	0.24	0.18			3.0	Bearings ran well.
E-47	B-5a	R-6a	8	5.5	500	0	174	89	0.24	0.18	0.014	0.014	2.5	Bearings ran well at maximum scheduled thrust load.
E-48	B-5a	R-6a	8	8 (18 max)	500	648	326 max	513	0.24	0.18	0.015	0.015	2.5	Same as E-47 with heated carrier air. Ball bearing temperature did not stabilize. Test stopped when there was a sudden jump in torque to 18 in-lb.
E-49	B-6	R-6b	8	4.0	0	0	94	98	0.24	0.15	0.015	0.015	2.0	Calibration run with new bearings.
E-50	B-6b	R-6b	8	5.0	500	0	164	82	0.24	0.16	0.013	0.013	2.0	Check results of run E-47. Bearings ran well.
E-51	B-6b	R-6b	12	18 max	500	0	350 max	96	0.24	0.10			0.5	Shut down when torque became excessive after 33 minutes. The two inner rings of the ball bearing were welded together at the race surface.
E-52	B-4	R-6b	8	4.4	0	0	113	106	0.24	0.18	0.012	0.013	2.1	Calibration run of new ball bearing design.
E-53	B-4	R-6b	8	5.0	50	0	145	97	0.24	0.18	0.015	0.012	2.3	Bearings ran well.
E-54	B-4	R-6b	8	4.7 (8.0 max)	100	0	168 (200 max)	98	0.24	0.18	0.013	0.012	0.9	Test stopped when ball bearing temperature rose rapidly and torque became excessive after 56 minutes.
E-55	B-5a	R-6b	8	4.1	0	0	187	107	0.24	0.18	0.013	0.013	2.0	Calibration with new ball bearing.
E-56	B-5a	R-6b	8	4.3	50	0	156	99	0.24	0.18	0.011	0.011	2.2	Bearings ran well.
E-57	B-5a	R-6b	5	4.0	100	0	143	94	0.24	0.18	0.011	0.011	1.6	Bearings ran well.
E-58	B-5a	R-6b	12	6.1	50	0	166	116	0.24	0.18	0.013	0.013	2.0	Bearings ran well.

*Other temperature normalized for 75° F room ambient temperature.

TABLE III. SUMMARY OF BEARING TESTS (Cont)

Test No.	Ball (Front) Bearing S/N	Roller (Rear) Bearing S/N	Speed (rpm x 10^{-3})	Avg Torque (in-lb)	Thrust Load Front Bearing (lb)	Rear Bearing Heating (watts)	Stabilized Temp (°) Ball Bearing (° F)	Stabilized Temp (°) Roller Bearing (° F)	Airflow Ball Bearing (lb/min)	Airflow Roller Bearing (lb/min)	Powder Flow Ball Bearing (gm/min)	Powder Flow Roller Bearing (gm/min)	Time of Run (hr)	Remarks
E-59	B-5e	R-3b	8	NA	0	0	108	96	0.24	0.18	0.010	0.010	2.0	Checkout new thrust air labyrinth and rear bearing. Bearing ran well.
E-60	B-5e	R-3b	8	4.8	50	0	148	90	0.24	0.18	0.011	0.011	2.0	Check with test E-56. Bearings ran well.
E-61	B-5e	R-3b	12	7.0	50	0	182	132	0.24	0.18	0.010	0.015	[illegible]	Check with test E-50. Bearings ran well.
E-62	B-5e	R-3b	16	12.7	60	0	191	116	0.24	0.18	0.010	0.010	2.7	Bearings ran well
E-63	B-5e	R-3b	8	13.0	0	0	~90	~97	0.24	0.18	0.012	0.012	1.0	Run to check torque with counterweighted turbine disc. Shut down because of excessive windage torque.
E-64	B-5e	R-3b	8	3.3	0	0	82	91		0.18	0.008	0.008	2.0 4.5	Checkout run with dummy turbine disc. Following this run a high speed checkout run of 4-1/2 hours duration was made at up to 20,000 rpm. Torque and sound levels indicated that the bearings were performing well during this run. Corrected air leakage around front bearing, giving more accurate airflow.
E-65	B-5e	R-3b	16	10.5 12.4	0 50	0	148 307	131 111		0.18	0.011	0.014	2.9	Check with test E-62. Ball bearing temperature very high when 50 pound thrust load applied. Shut down.
E-66	B-5e	R-3b	8	3.5	0	0	106	93		0.18	0.010	0.010	1.2	Checkout of instrumentation. Bearings ran well.
E-67	B-5e	R-3b	8	3.4	50	0	135	84		0.18	0.010	0.010	1.0	Checkout of instrumentation. Bearings ran well
E-68	B-5e	R-3b	16	12.4 12.2	50 50	0 0	309 270	129 125		0.18 0.18	0.010 0.033	0.010 0.033	1.0 1.3	Check with test E-65. Increasing powder flow after 1 hour lowered bearing temperature. Wear tracks worn into outer ring of ball bearing by retainer.
E-69	B-5	R-3b	8	3.4	0	0	82	85		0.18	0.012	0.012	1.7	Checkout with bearing B-5. Bearings ran well.
E-70	B-5	R-3b	8	4.3	50	0	163	92		0.18	0.011	0.011	2.7	Shut down because of excessive front bearing noise.
E-71	B-5	R-3b	8	4.5	0 50	0 0	111 165	94 93		0.18 0.18	0.012 0.012	0.012 0.012	1.0 0.7	Thrust load applied after 1 hour running caused rapid 50° F increase in ball bearing temperature. Shut down because of noisy front bearing. Subsequent inspection revealed bearing mounts out of concentric by about 0.015 inch.
E-72	B-3a	R-1a	8	3.1 3.4	0 50	0 0	111 122	98 88		0.18 0.18	0.012 0.013	0.012 0.013	1.7 0.8	Checkout of realigned engine. Bearings ran well before and after thrust load was applied at 100 minutes
E-73	B-3a	R-1a	12	6.3 6.9	0 50	0 0	119 151	109 101		0.18 0.18	0.012 0.012	0.012 0.012	1.6 1.0	Comparison with test E-61. Bearings ran well. (Note: E-61 was with different bearing design.)
E-74	B-3a	R-1a	16	10.1 10.6	0 50	0 0	139 180	119 120		0.18 0.18	0.012 0.012	0.012 0.012	1.0 1.0	Comparison with test E-62. Bearings ran well.
E-75	B-3a	R-1a	20	15.2	0 50 100	0 0 0	182 195 263 Max	139 138 134		0.18 0.18 0.15	0.020 0.020 0.020	0.020 0.020 0.020	1.2 1.0 0.5	Successful completion of testing of B-3 ball bearing design with 50 pound thrust load. Ball bearing temperature increased rapidly and bearing became noisy when thrust load was increased to 100 pounds after 2.2 hours. Ball bearing temperature did not stabilize with this load.
E-76	B-5d	R-1a	8	3.2 3.7	0 50	0 0	118 130	101 94		0.18 0.18	0.015 0.015	0.015 0.015	1.5 1.0	Checkout and break in of new ball bearing. Bearings ran well. (No grooves in ball bearing retainers.)

TABLE III. SUMMARY OF BEARING TESTS (Cont)

Test No.	Ball (Front) Bearing S/N	Roller (Rear) Bearing S/N	Speed (rpm x 10^{-3})	Avg Torque (in-lb)	Thrust Load Front Bearing (lb)	Rear Bearing Heating (watts)	Stabilized Temp (°)* Ball Bearing (°F)	Stabilized Temp (°)* Roller Bearing (°F)	Airflow Ball Bearing (lb/min)	Airflow Roller Bearing (lb/min)	Powder Flow Ball Bearing (gm/min)	Powder Flow Roller Bearing (gm/min)	Time of Run (hr)	Remarks
E-77	B-4a	R-4	8	3.9	50	0	141	84		0.18	0.012	0.012	2.0	Check of rebalanced main rotor shaft with new bearings. Bearings ran well.
E-78	B-5d	R-4	8	3.6	70	0	136	91		0.18	0.012	0.012	2.0	Check with test E-75. Bearings ran well
E-79	B-5d	R-4	13	[illegible]	50	0	103	95		0.18	0.013	0.013	2.0	Bearings ran well but seemed noisier than usual. Next run started at 17,000 rpm was shut down because of excessive noise.
E-80	B-5d	R-4	5	3.3	0	0	111	106		0.18	0.015	0.015	2.0	Grooves cut in B-5d retainer. Bearings ran well
E-81	B-5d	R-4	16	12.3	0	0	250 Max	NA					2.0	Difficulties with powder lubricator invalidated results of this test.
E-82	B-5d	R-4	12	6.7	0	0	137 Min 126 Max	113 Min 120 Max		0.18	Varied		6.3	Powder flow rate varied by increasing lubricator feedwheel speed from the normal of 2 rpm to 10 rpm. There was about a 5° F drop in ball bearing temperature at high feedwheel speeds compared to that at 2 rpm, and about a 5° F increase in roller bearing temperature as the feedwheel speed was increased.
E-83	B-5d	R-4	16	12.9 10.2	0 0	0 0	279 135	138 145		0.18 0.18	4 rpm 5 & 6 rpm	2 rpm 4 & 6 rpm	2.5 4.3	Ball bearing temperature dropped substantially when lubricator feedwheel speed was increased from the normal speed of 2 to 4 rpm. There was no appreciable change when feedwheel speed was further increased to 6 rpm.
E-84	B-5d	R-4	16	15.0 16.2	0 50	0 0	163 198 Max	175 133		0.18 0.18	4 & 6 rpm 4 rpm	4 & 6 rpm 3 rpm	1.9 1.0	Thrust load applied after 1.9 hours. At this time ball bearing temperature increased and restabilized at 198° F. After 30 minutes, the ball bearing temperature dropped and restabilized at 182° F
E-85	B-5d	R-4	20	14.7	50	0	~260	NA		0.18	4 & 6 rpm	4 & 8 rpm	2.8	Heated carrier air. Ball bearing temperature stabilized at 170° F after 90 minutes. After 20 minutes at this level, the temperature increased and restabilized at about 260° F but fluctuated several degrees around this value. This test marked the successful completion of testing of the B-5 series bearing with 50 pounds thrust load.
E-86	B-5d	R-4	20	15.9	78	0	306	135 Max		0.18	4 & 8 rpm	4 & 8 rpm	3.8	Ball bearing temperature stabilized at 306° F with feedwheel speed at 4 rpm. When feedwheel speed was increased to 8 rpm, ball bearing temperature started to drop and appeared to be stabilizing to about 270° F. Roller bearing temperature varied between 140° F and 140° F.
E-87	B-5d	R-4	20	16.8	66	0	280	90		0.18	0.025	0.025	2.0	Started with thrust load of 130 pounds which dropped to 66 pounds. Bearings ran well.

*Temperature normalized for 75° F room ambient temperature.

TABLE IV. SUMMARY OF PREVIOUS ENGINE TESTING, UNFIRED J-69 ENGINE

4

Test No.	Ball (front) Bearing S/N	Roller (rear) Bearing S/N	Speed (rpm x 10^{-3})	Avg Torque (in-lb)	Thrust load Front Bearing (lbs)	Rear Bearing Heating (watts)	Stabilized Temp (°) Ball Bearing (° F)	Stabilized Temp (°) Roller Bearing (° F)	Air Flow Ball Bearing (lb/min)	Air Flow Roller Bearing (lb/min)	Powder Flow Ball Bearing (gm/min)	Powder Flow Roller Bearing (gm/min)	Time of Run (hrs)	Remarks
E1	B-1c	R-7	2		0	0	77	93	0.15	0.15			2.0	Both brgs as supplied with engine. Ball bearing with grooved retainer. Bearing material 52100 steel - both brgs. Rig checkout. Total powder flow = 0.008 gm/min.
E2	B-1c	R-7	2		0	0	75	73	0.15	0.15			1.0	Rig checkout. Total powder flow = 0.005 gm/min.
E3	B-1c	R-7	2		0	0	92 max	118 max	0.15	0.14			1.0	Rig checkout. Temperature did not stabilize. Total powder flow = 0.018 gm/min.
E4	B-1c	R-7	6	3.6	0	0	85	116 max	0.15	0.14			2.0	Rig checkout. Roller bearing temperature did not stabilize. Total powder flow = 0.046 gm/min.
E5	B-1c	R-7	6	2.7	0	0	80	154	0.15	0.14			3.0	First low speed test. Total powder flow = 0.044 gm/min.
E6	B-1c	R-7	8	4.8 (5.7 max)	0	0	93 max	125 max	0.15	0.14	0.005	0.039	1.0	Tube system checkout. Temperature did not stabilize.
E7	B-3	R-3a	0.5	0.4	0	0	74	74	0.15	0.14			1.0	New bearings. Ball bearing of M50 steel. Roller bearing of 440C stainless steel. Ball bearing with ground retainer. Run to coat bearings.
E8	B-3	R-3a	2	0.8	0	0	76	78	0.15	0.14			1.5	Total powder flow = 0.024 gm/min.
E8a	B-3	R-3a	8	3.9	0	0	90	90	0.15	0.14			1.75	Total powder flow = 0.024 gm/min.
E9	B-3	R-3a	8	3.6	0	0	86 max	88 max	0.15	0.14			1.0	Tube system checkout. Temperature did not stabilize. Total powder flow = 0.038 gm/min.
E10	B-3	R-3a	8	3.6	0	0	90	89					3.0	Ball brg temperature rose to 94° F at end of test. Total powder flow = 0.024 gm/min.
E11	B-3	R-3a	8	3.8	0	0	98	97					2.3	Bearing temperature fluctuated around stabilization values. Total powder flow = 0.028 gm/min.
E12	B-3	R-3a	8	4.0	0	0	91	87			0.008	0.011	2.0	Lube system checkout. Bearings ran well.
E13	B-3	R-3a	12	8.0	0	0	112	104			0.010	0.010	1.3	Bearings ran well.
E14	B-3	R-3a	12	8.0	0	0	111	104			0.010	0.010	1.0	Check on run E13. Bearings ran well.
E15 to E19	B-3	R-3a	8-11		0	0							1.7	Rig checkout. About 1.7 hours total running time.
E20	B-3	R-3a	8	3.5	0	0	101	97	0.76	0.32	0.009	0.010	3.0	New lubricant distribution system. Bearings ran well.
E21	B-3	R-3a	8	3.7	0	0	96	90	0.76	0.32	0.011	0.003	2.3	Bearings ran well.
E22	B-3	R-3a	8	3.5	0	70	99	153	0.76	0.32	0.010	0.008	3.0	Total heater capacity = 6 x 600 watts plus 1 x 1100 watts. Bearing ran well.
E23	B-3	R-3a	8	3.6	0	172	98	229	0.76	0.32	0.016	0.017	2.0	Bearings ran well.
E24	B-3	R-3a	8	3.5	0	259	99	300	0.76	0.32	0.016	0.017	2.3	Bearings ran well.
E25	B-3	R-3a	8	3.4	0	460	104	430	0.76	0.31	0.013	0.013	4.0	Bearings ran well.
E26	B-5	R-6	0.5	1.1	0	0	76	75	0.24	0.18	0.023	0.023	1.0	New bearings. Ball bearing of 440CM modified stainless steel with grooved retainer. Roller bearing of M50 tool steel. New lube system. Run to coat bearings.
E27	B-5	R-6	8	3.6	0	0	91	91	0.24	0.18	0.023	0.023	3.0	Calibration at 8000 rpm. Bearings ran well.
E28	B-5	R-6	8	3.7	0	0			0.24	0.18	0.013	0.013	0.7	Shut down to repair test rig.
E29	B-5	R-6	8	3.9	0	0	93	92	0.24	0.18	0.013	0.013	3.0	Bearings ran well.
E30	B-5	R-6	8	3.8	0	253	98	252	0.24	0.18	0.014	0.014	1.7	New heating arrangement. Total heater capacity = 6 x 600 watts plus 1 x 120 watts.
E31	B-5	R-6	8	3.8	0	385	101	342	0.24	0.18	0.014	0.014	2.3	Bearings ran well.
E32	B-5	R-6	8	3.4	0	648	101	518	0.24	0.18	0.011	0.011	4.0	Operating at expected temperature level of rear bearing in fired engine.
E33	B-5	R-6	8	3.3	0	648	128	540	0.24	0.18	0.010	0.010	2.3	THIS AND ALL SUBSEQUENT TESTS MADE WITH HEATED CARRIER AIR UNLESS OTHERWISE NOTED. AIR TEMPERATURE AT BEARING INLETS KEPT AT ABOUT 250° F. BEARINGS RAN WELL.
E34	B-5	R-6	12	6.6	0	648	143	435	0.24	0.18	0.011	0.011	3.3	Bearings ran well. Heated rear bearing running cooler than at 8000 rpm.
E35	B-6a	R-6a	8	3.9	0	0	98	97	0.24	0.18	0.009	0.009	2.0	New bearings. Calibration run as a check with test E29. Unheated carrier air.
E36	B-6a	R-6a	8	3.6	0	648	130	325	0.24	0.18	0.009	0.009	2.0	Run to check rear bearing cooling with modified ducting in engine (test stand application only). Bearings ran well.
E37	B-5a	R-6a	12	6.2	0	648	148	272	0.24	0.18	0.013	0.013	[illegible]	Same as above, at 12,000 rpm. Bearings ran well.
E38	B-5a	R-6a	12	6.6	0	645	155	424	0.24	0.18	0.011	0.011	2.0	Original ducting as for tests E1 to E35. Calibration check with test E34. Bearings ran well.

*Outer race temperature normalized for 75° F room ambient temperature

PHASE 3 ENGINE TESTING (TEMPERATURE AND TORQUE VERSUS THRUST LOAD)

Phase 3 testing consisted of running the unfired J-69 engine at 8,000 rpm, with unheated carrier air and unheated rear bearing. Thrust loading on the front bearing was varied from 0 to 500 pounds. Figure 29 indicates the results of the temperature and torque versus thrust load tests using the bearing series B-5a and R-6a. As the thrust load was increased the temperature rose fairly rapidly, leveled off at 100 pound load, remained steady until at 250 pound load the temperature again started to climb to a maximum value of 170° F at a 500 pound load. An attempt was made to operate at 12,000 rpm with a 500 pound load, but the test was unsuccessful as the bearing temperature reached 350° F after 33 minutes of operation indicating a failure of the bearing. Detailed information and pictures of the failure are described in test E-48 in the Appendix.

Although ball bearings performed well at all loads, there was about a 40 percent increase in torque required to drive the engine as load was increased from 0 to 500 pounds. The torque curve basically follows the slope of the temperature curve.

Total accumulated running time of the bearing series B-5a and R-6a during the Phase 2 and Phase 3 tests amounted to 31.3 hours before failure of the B-5a bearing recurred.

PHASE 4 ENGINE TESTING (TEMPERATURE AND TORQUE VERSUS SPEED)

Phase 4 testing consisted of running the unfired J-69 engine at varying speeds from 8,000 to 20,000 rpm and at 0 and 50 pound thrust load with heated carrier air and unheated rear bearing. Figure 30 indicates the results of the temperature and torque versus speed tests using the bearing series B-3a and R-1a. Figure 31 indicates similar results of the tests using the bearing series B-5c and R-3b.

The curves shown on figure 30 (B-3a and R-1a) are representative of the characteristic performance of a given set of bearings. In as much, as a few of the plotted test points on figure 31 are taken from other bearing configurations the curves as presented are not completely valid, although the data confirms the results shown in figure 31.

The torque curves at 0 and 50 pound loads are practically parallel to each other throughout the speed range. At a given speed the gap between the two curves is reasonably constant within a range of 7 to 10 percent. The average increase of torque to drive the engine, regardless of the load, increased about 450 percent as the speed increased from 8,000 to 20,000 rpm. The temperature change as the speed was varied from 8,000 to 20,000 rpm increased about 48 percent at the 0 thrust load condition, to 56 percent for the 50 pound thrust load condition.

The bearing series B-3a and R-1a were successfully run for a period of 9 hours before the failure of the B-3a bearing occurred while attempting to operate at 20,000 rpm and a thrust load of 100 pounds.

SUMMARY

- A rear (roller) bearing was successfully run at 20,000 rpm and 515° F stabilized operating temperature, simulating estimated engine operating conditions. The bearing used had inner ring guided rollers and an outer

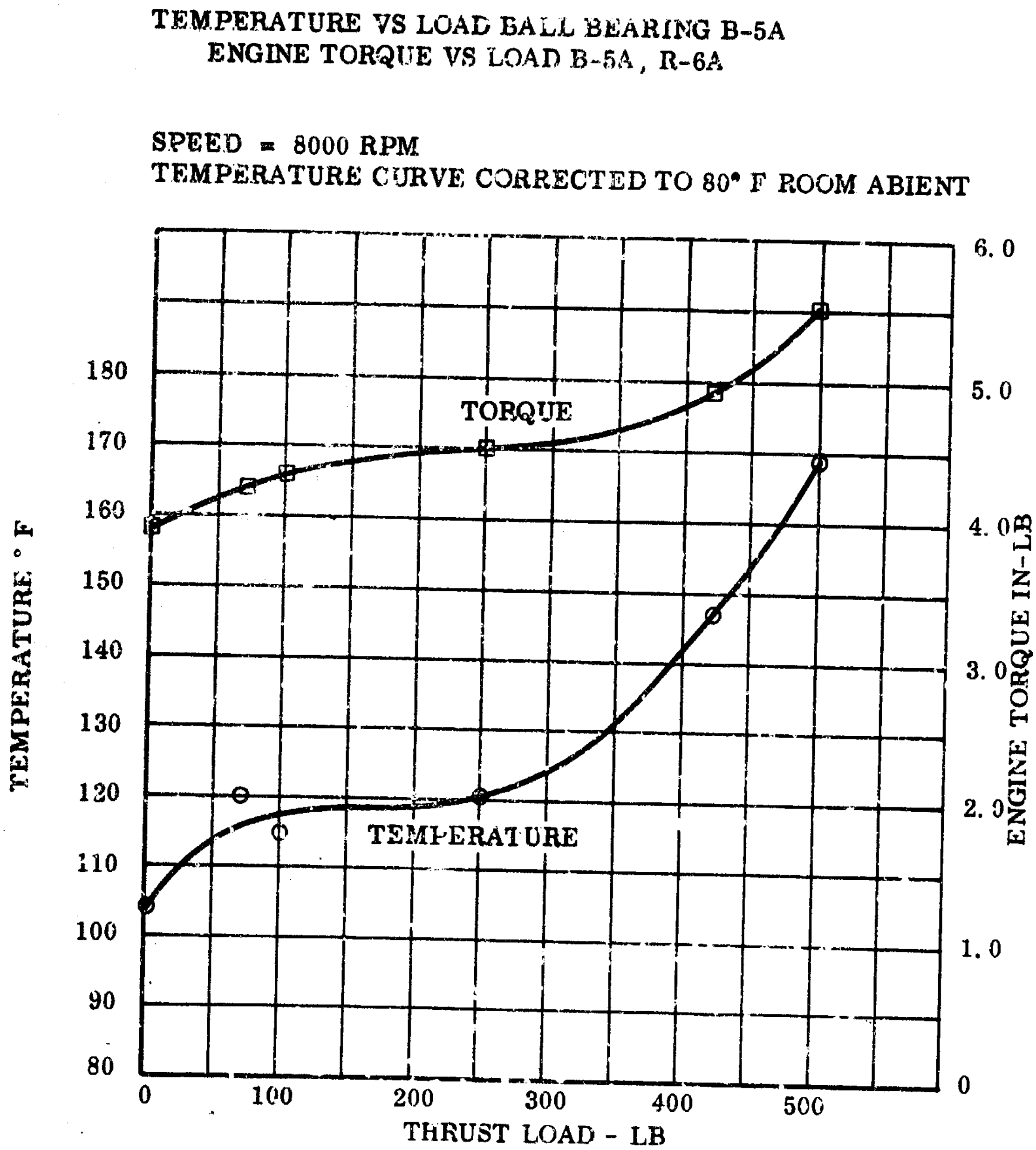


Figure 29. Phase 3 Test Results, Temperature and Torque Versus Thrust Load

TEMPERATURE VS SPEED AT 0 AND 50 LB THRUST LOAD (BALL BEARING B-3a)
ENGINE TORQUE VS SPEED (B-3a R-1a)
ALL TEMPERATURES CORRECTED FOR 75° F ROOM AMBIENT
POWDER FLOW = 0.012 TO 0.020 GRAM/MIN EACH BEARING
ALL TESTS MADE WITH CARRIER AIR TEMPERATURE OF 220° F

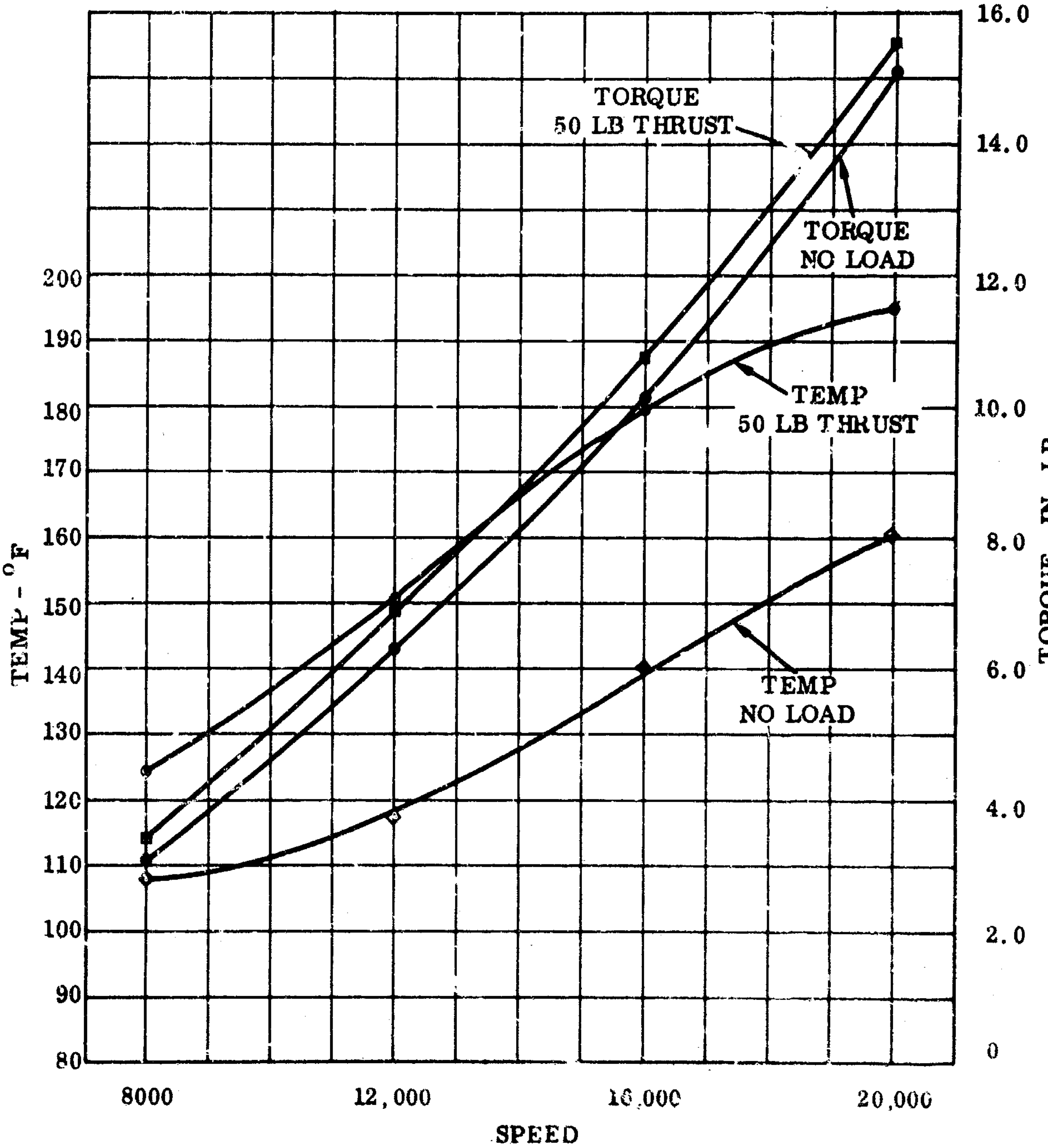


Figure 30. Phase 5 Test Results, Temperature Versus Speed and Load, Bearing Series B-3

TEMPERATURE VS SPEED @ 50 LB THRUST LOAD (BALL BEARING BVc) ENGINE
TORQUE VS SPEED (BVc, RIII6)
COMPARISON WITH NO LOAD PINS (BV, RVI)
ALL TEMPERATURES CORRECTED FOR 75° F ROOM AMBIENT
POINTS MARKED BVd WERE FOR APPROXIMATE STABILIZATION ACHIEVED WITH BEARING BVd, RIVa

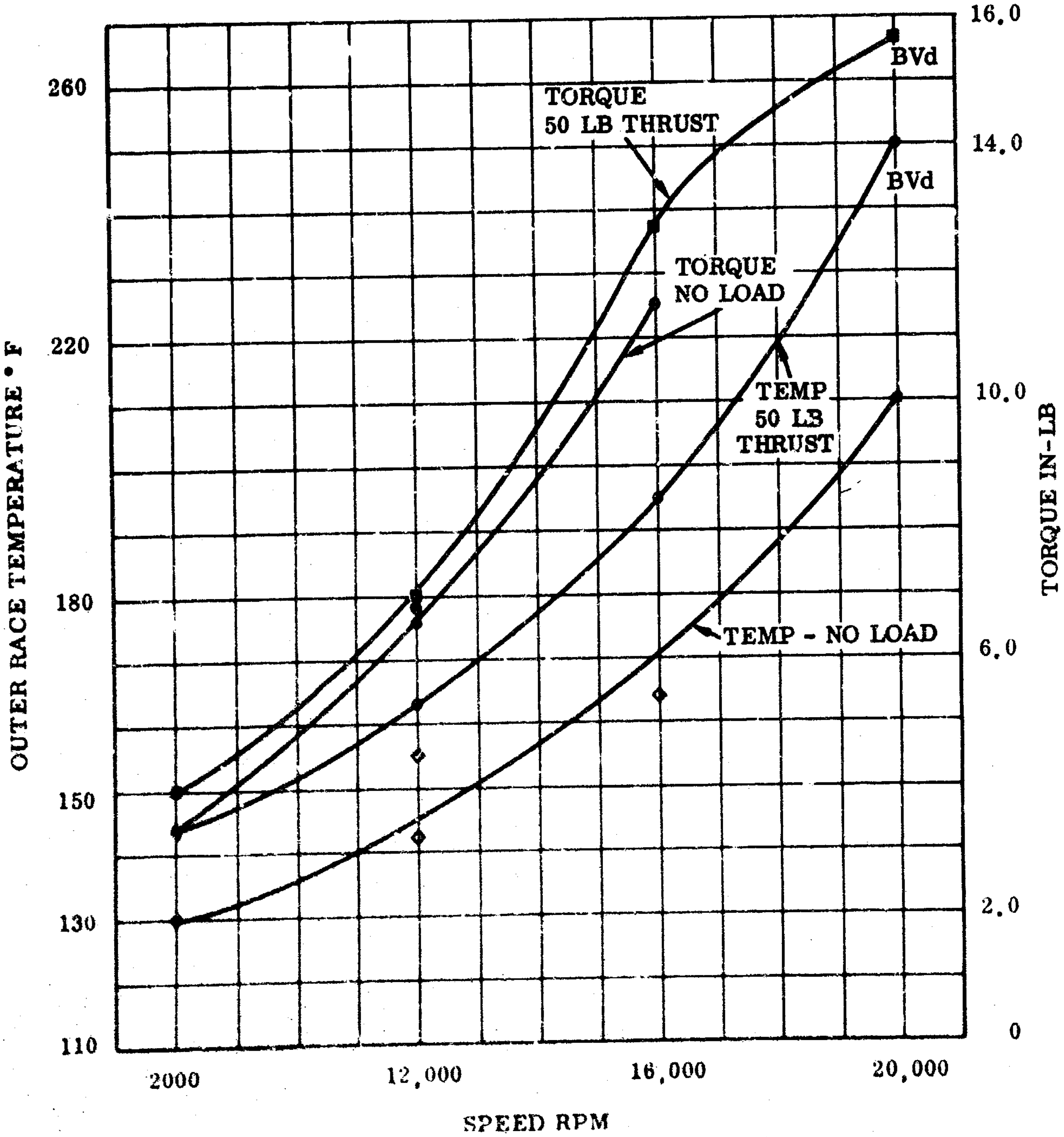


Figure 31. Phase 5 Test Results, Temperature Versus Speed and Load Bearing Series B-5

ring guided retainer. Rings and rollers are of M-50 tool steel and the retainer is Monel S.

- A ball bearing was run successfully at 8,000 rpm and 500 pounds thrust load (50 pounds radial load). This bearing was of conventional split inner ring design with an outer ring guided retainer. The retainer was modified by cutting grooves in the outer rim of the lubricant exhaust side to facilitate lubricant flow through the bearing. Rings and balls were 440 CM stainless, and the retainer was silver plated AMS 6415 steel.

- Testing at simulated front bearing operating conditions of 50 lbs thrust load and 20,000 rpm with heated (bleed air temperature) carrier air was successfully performed with two different bearings of different designs. (B3 and B5), described in section IV.

- A ball bearing was successfully operated at 20,000 rpm with heated carrier air and 90 - 100 lbs thrust load.

- Testing revealed that optimum lubricant powder flow for ball bearing operation under thrust loaded conditions is about 0.015 grams per minute. This is approximately double the flow rate which had been used for most earlier testing.

- The increase in powder flow rate appeared to have little if any affect on a roller bearing operating at 20,000 rpm with no external heating.

SECTION XI

FIRED ENGINE DESIGN

GENERAL

The turbojet engine used as the test engine for this program is a USAF Model J-69-T-25, manufactured by the Continental Aviation and Engineering Corporation. This engine is shown in Figure 1 with accessories mounted in place and the conventional oil lubrication system installed. A cutaway drawing showing an identical engine with a powder lubrication system installed is shown in figure 32. This drawing depicts the powder - air flow as well as the ejection air flow lines.

FIRED ENGINE LUBRICATION SYSTEM DESIGN

Figure 33 shows the J-69 engine with modifications and lube system flow paths for operation with powder lubricated engine main bearings.

In a normally (oil) lubricated engine, there is a face type oil seal on the rotor shaft immediately behind the front bearing. For the powder lubricated engine this seal has been relocated at the gearshaft end of the accessory housing, as shown in figure 34. In this position the face seal will serve as the primary means of isolating the front main bearing (powder lubricated) from the oil used in the accessory gearbox. Complete isolation of the powder system from any oil contamination is necessary for both theoretical and practical reasons. Theoretically, because the object of the test program is to determine the effectiveness of an absolutely dry lubrication system; practically, because a small amount of oil mixed with the graphite might cause severe caking of the powder in the bearings. To provide for the complete lubrication system isolation required, a slinger is provided as shown in figure 34. This slinger will induce any oil which leaked through the face seal to be ejected directly into the exhaust air stream from the front bearing, thus preventing the oil from getting to the bearing itself.

At the front bearing, the air-lubricant mixture will be carried through passages drilled in the 10 o'clock strut of the compressor housing. The powder is then routed past the birdcage bearing support cage and directed into the bearing. While some rework of the support cage was necessary, the basic geometry and tolerances were maintained to meet critical speed characteristics of the main rotor shaft. A boost air ejector is provided in the powder flow system to overcome the pressure drop and prevent powder buildup at the transition from the compressor housing to the support cage. After passing through the front bearing, the powder is picked up by a slinger (replacing the original engine oil seal) and is flung out, exhausting into the engine inlet air stream.

Figure 35 shows the detail parts of the front bearing assembly. The thermocouple wires shown, are installed to monitor the temperature of the outer race of the bearing. The segment shown at the bottom of the bird cage fits into a cutaway of the front engine housing shown in figure 36. The bearing depicted in figure 35 is shown with the modified retainer, slinger, lock nut and spacers.

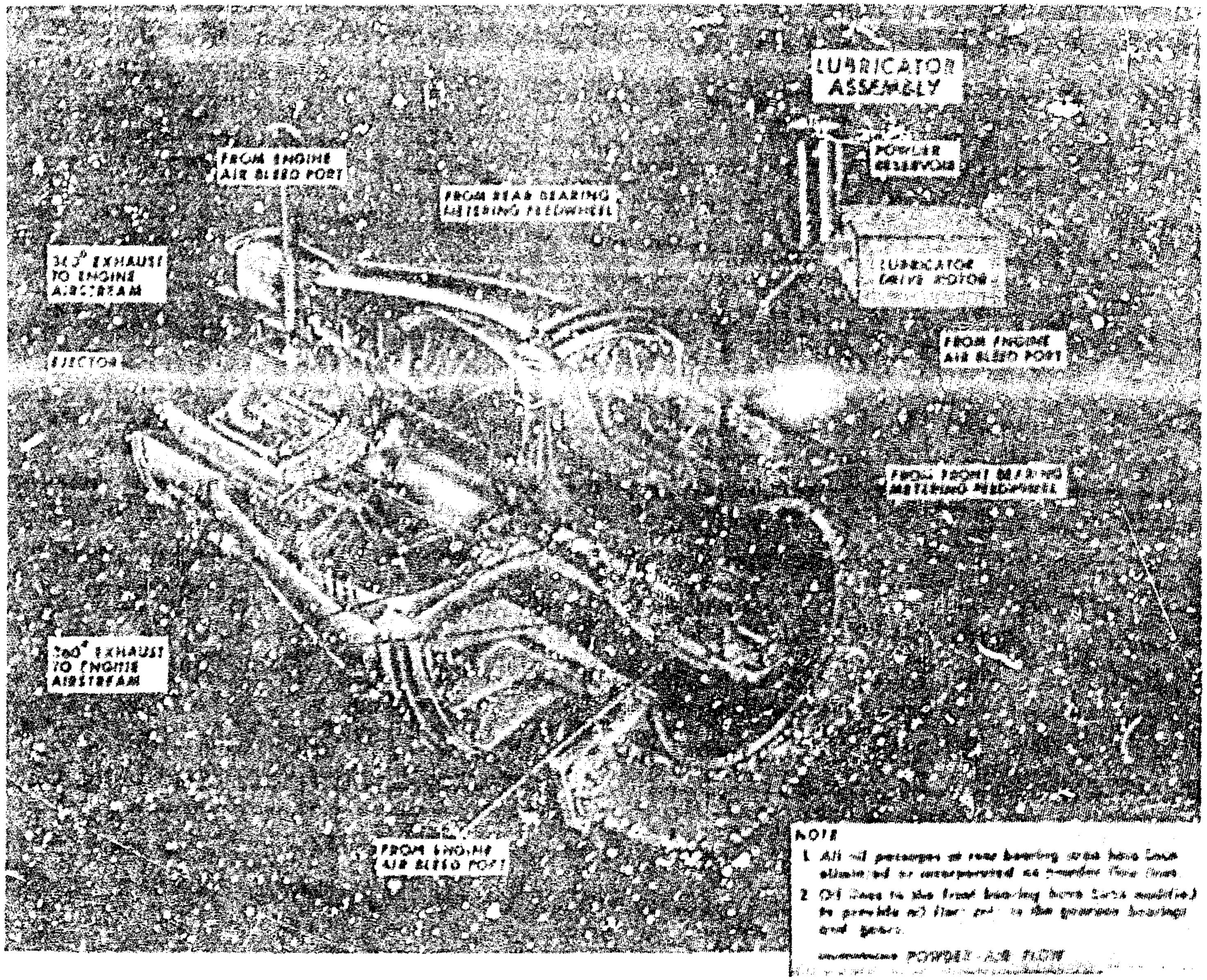


Figure 32. I-69-T-25 Modified for Powder Lubrication Cutaway View

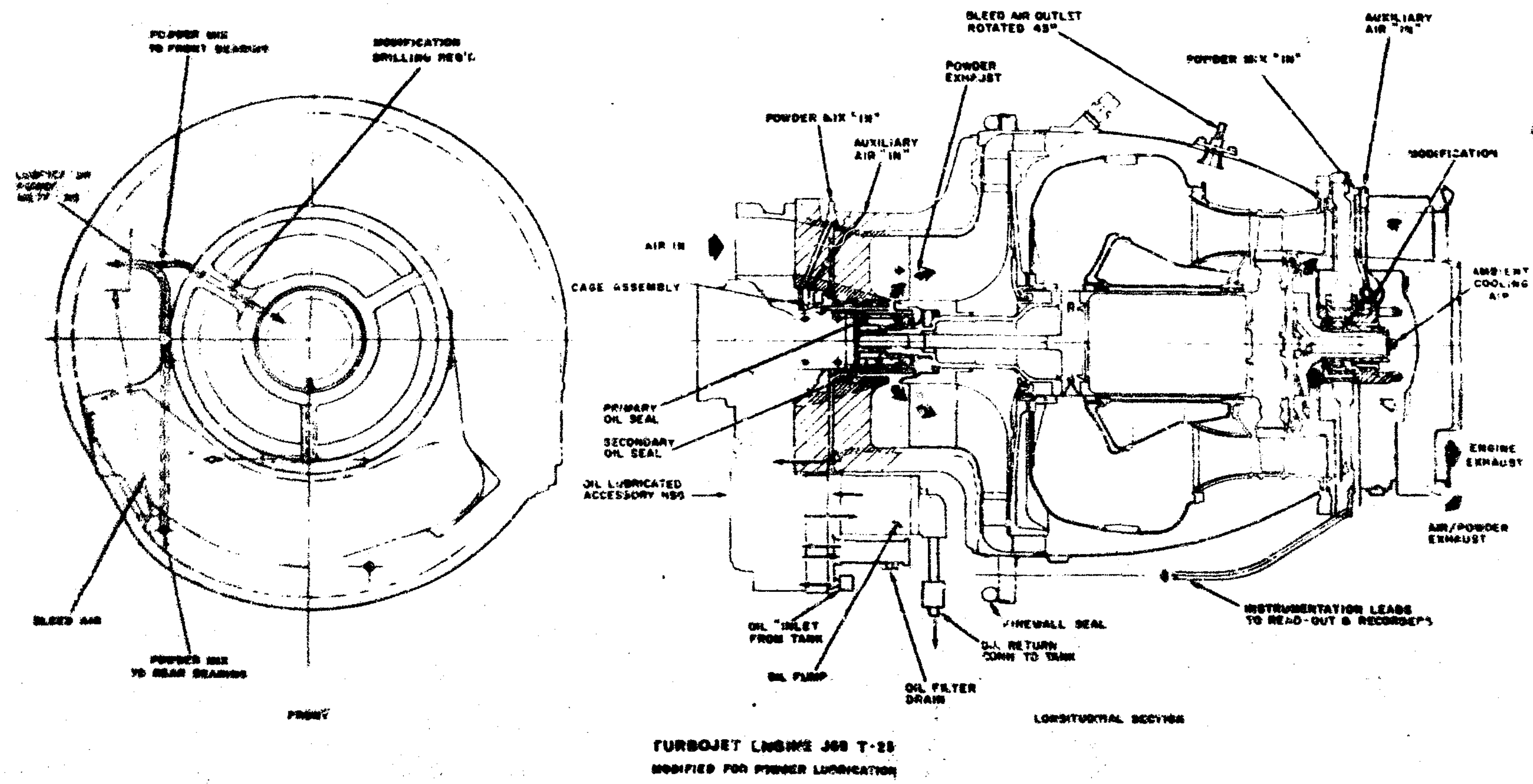


Figure 33. J-69-T-25 Modified for Powder Lubrication

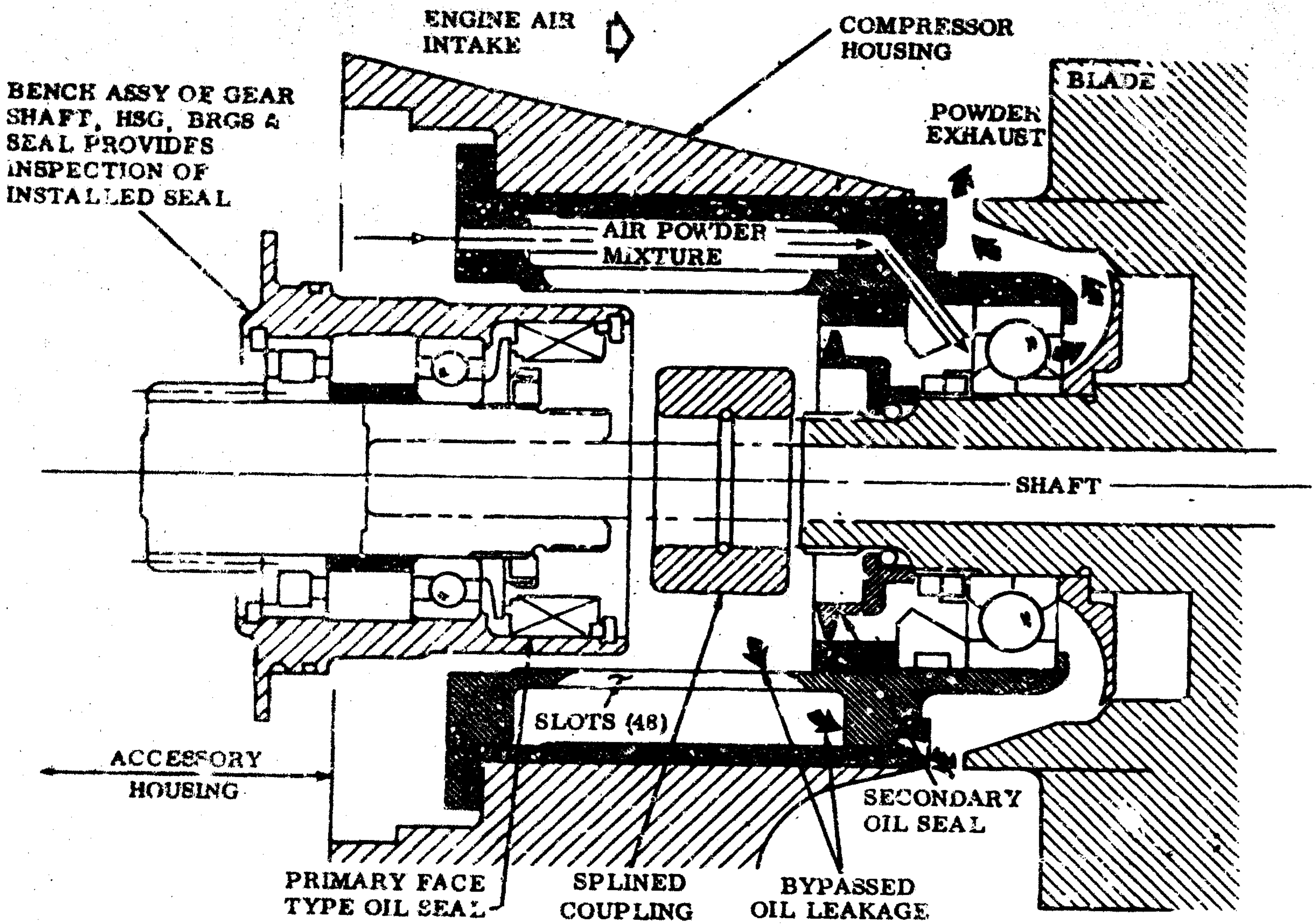


Figure 34. Front Bearing Modified for Powder Lubrication

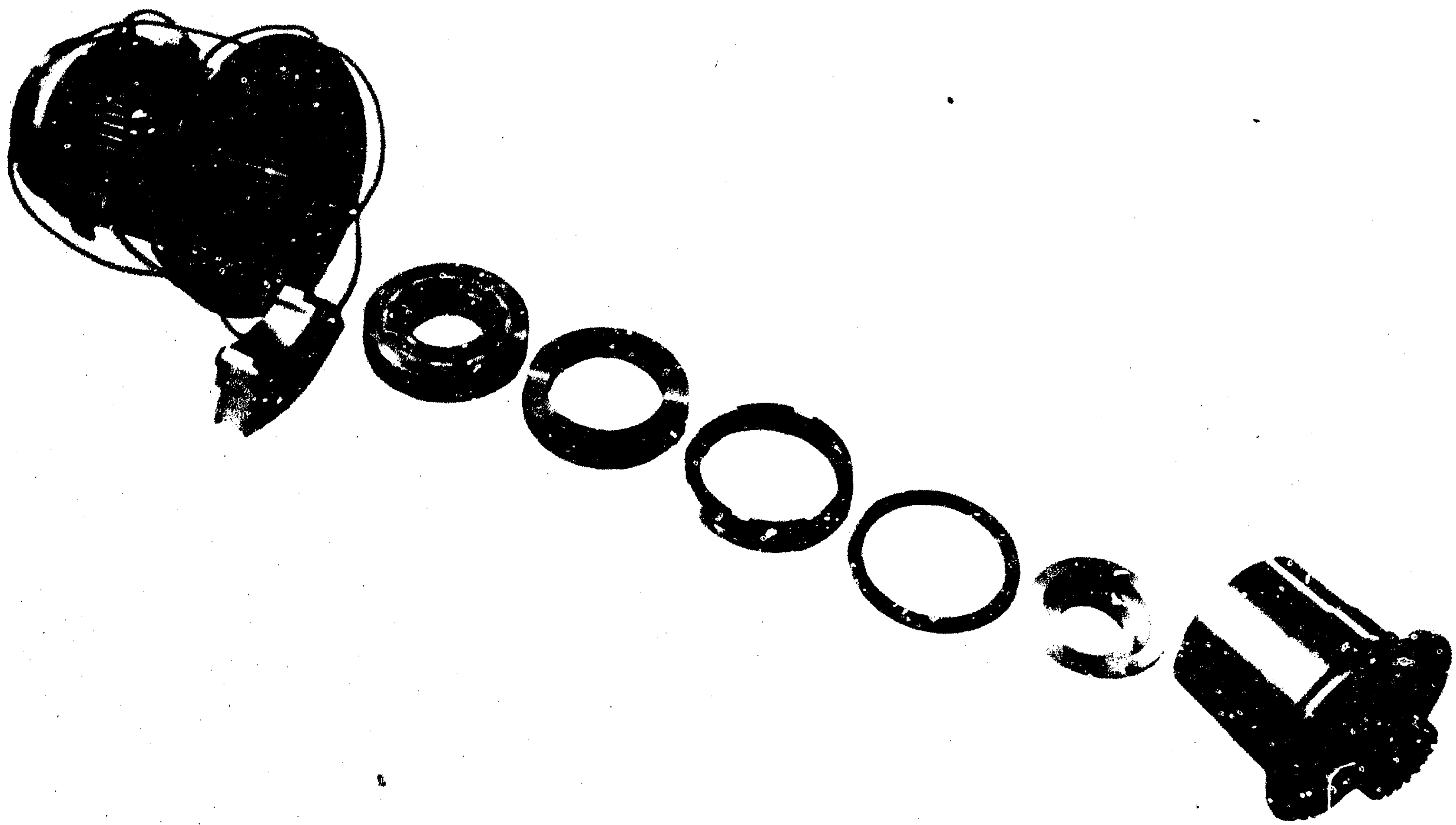


Figure 35. Front Bearing Assembly Detail

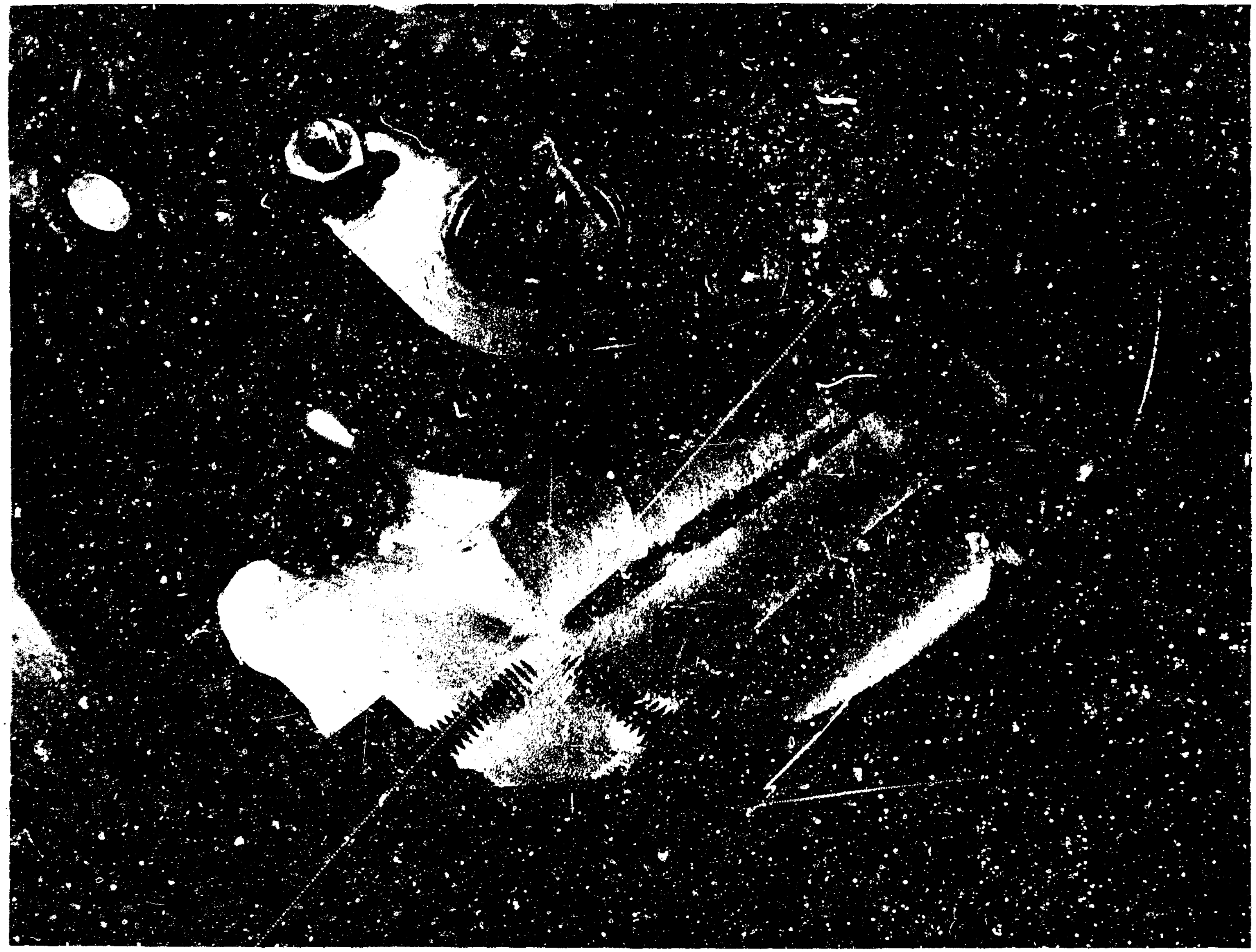


Figure 36. Front Bearing Housing Modification

At the rear bearing, the powder flow path can be readily adapted from the existing oil system plumbing with only minor modifications. The air-powder mixture flows through piping in a strut similar to that used in the oil lubricated engine. After flowing through the bearing, the powder is thrown out of the bearing area to ambient air by means of a slinger which replaces the original stepped labyrinth at the front end of the rear bearing housing.

The rear bearing assembly (figure 37) is shown with the thermocouple wire around the rear bearing housing. The bearing slinger, bearing, and bearing ejector housing are also shown.

Figure 38 is a detail view of the ejector housing and the labyrinth seal.

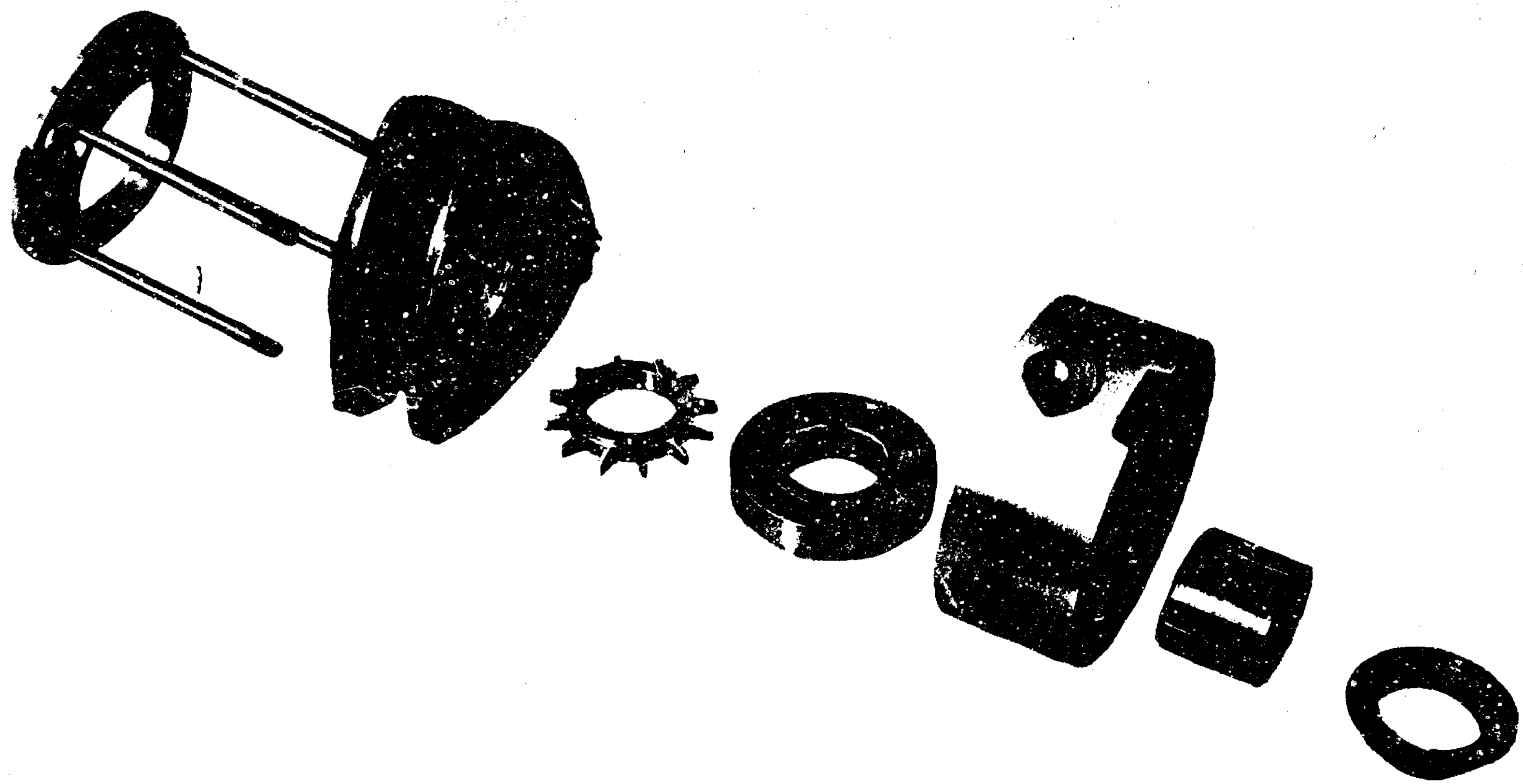


Figure 37. Rear Bearing Assembly Detail

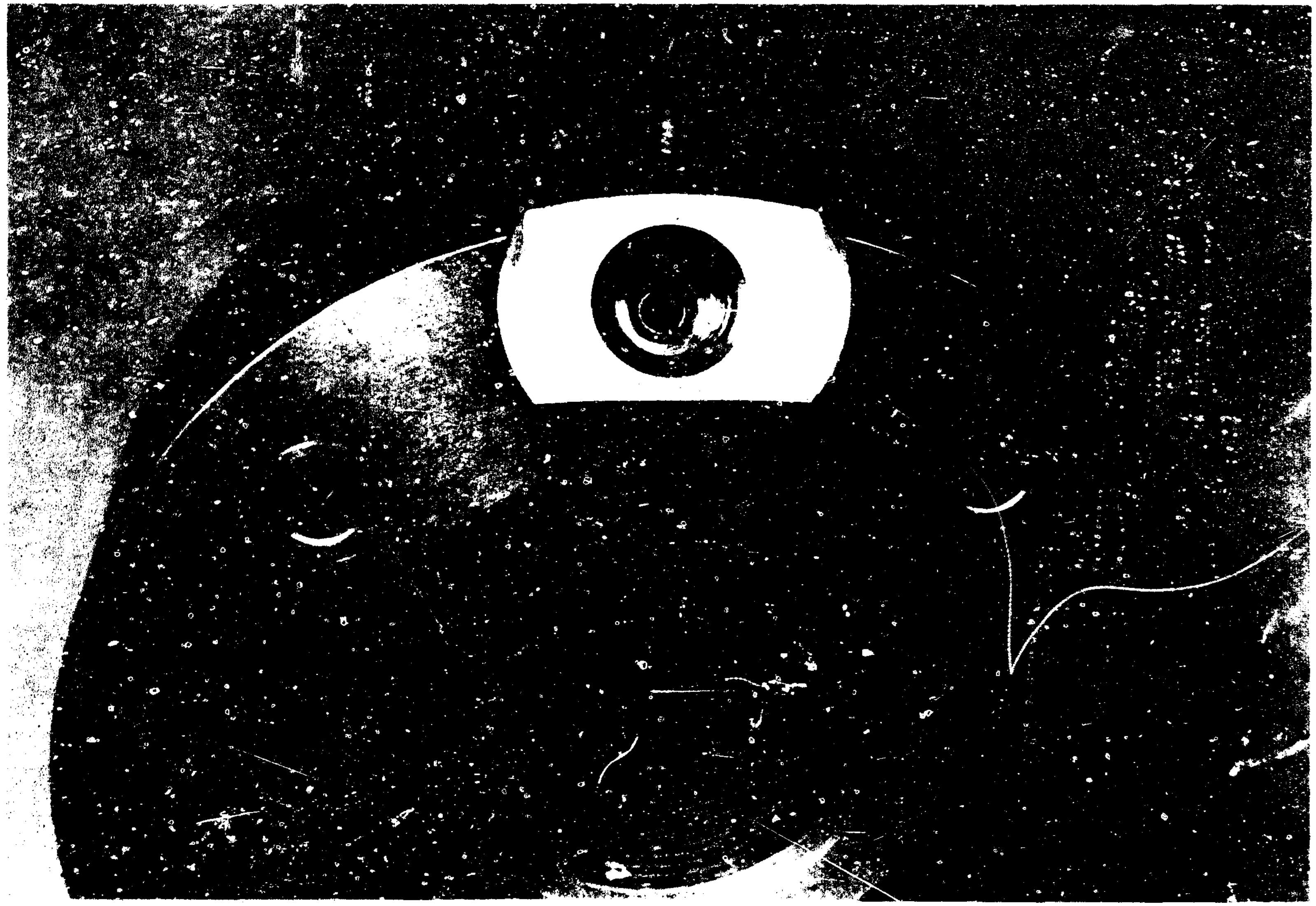


Figure 38. Rear Bearing Ejector Housing

SECTION XII

CONCLUSIONS AND RECOMMENDATIONS

The original objective of the three year program, concluded with this report, was the development of a powder lubrication system, and bearings, to be used with a J-69 turbojet engine running under fired conditions on a test stand. This development program was successfully completed with the unfired testing of a J-69 engine using such a system under simulated fired engine operating conditions.

Major achievement during this program included the following:

- Operation of a roller bearing (engine rear bearing) design with powder lubrication at 20,000 RPM and 540° F operating temperature, simulating conditions expected in a fired J-69 engine with powder lubrication.

- Operation of two different ball bearing (engine front bearing) designs with powder lubrication at 20,000 RPM and 50 to 100 pounds thrust load, simulating estimated fired engine conditions with powder lubrication.

- Redesign of the J-69 engine lubrication system for use with powder lubrication under fired engine conditions.

- Modification was made to a fired J-69 engine in accordance with the above design changes. This engine has been shipped to Wright-Patterson Air Force Base, for actual fired testing with powder lubrication.

CONCLUSIONS

As a result of testing under the program reported, it appears that a J-69 engine can be successfully operated with powder lubrication replacing the normal oil lubrication system for the two main rotor shaft bearings. A preliminary design for the powder lubricating system has been successfully fabricated and tested on an unfired J-69 engine. For use with this system, the normal rear bearing of 52100 steel is replaced with one of M50 tool steel to provide for the high operating temperatures expected. The front bearing has been modified by the addition of air powder scavenge grooves cut in the retainer. The front bearing is made of 440 CM stainless steel to provide a safety factor against any unexpected thermal conditions occurring in the actual engine with powder lubrication.

A J-69 engine modified for use with powder lubrication was fitted with these bearings and shipped to Wright-Patterson Air Force Base for testing under actual fired conditions. The purpose of this future testing is to substantiate results already obtained with an unfired engine, as well as to determine any further bearing developments needed in a fired engine using powder lubrication.

RECOMMENDATIONS

As a result of the tests conducted during this program the following recommendations for further deployment of powder lubrication systems can be considered.

1. Use of powder lubrication as a primary lubrication system for turbojet engines when bearing temperatures exceeds the limitations of conventional lubricants.

2. Use of powder lubrication as an emergency lubrication system for engines, transmissions or auxiliary power units.

3. Use of powder lubrication as a primary lubrication system for rotating equipment exposed to a radioactive environment.

4. Commercial application for powder lubrication systems in steel mills, kilns and high temperature environments.

5. Use of powder lubrication in space environments where high bearing loads, and/or temperature and space vacuum dictate the replacement of a conventional lubrication system.

SECTION XIII

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SECTION XIV

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APPENDIX I

DESCRIPTION OF TEST RUNS

This section summarizes all of the significant test results obtained during this report period with specific emphasis on those tests which achieved the program's objectives.

Test E-39 was run at 12,000 rpm with a heated rear bearing as a check with run E-38, which was the last test made before the new drive system was installed. Rear bearing stabilization temperature was about 470° F, which was 50° F higher than the stabilization achieved in run E-38, but this difference was caused by test E-39 being run at 703 watts heating instead of the required 648 watts.

Test E-40 was run at 16,000 rpm with rear bearing heating starting at 703 watts, which was reduced to 648 watts with a corresponding temperature reduction of about 50° F, confirming the cause of the difference in temperature between tests E-38 and E-39. The rear bearing operated satisfactorily for 4 hours at about 465° F. Curves of race temperature versus time are shown in figure 39.

Test E-41 was run at 20,000 rpm, and rear bearing temperature stabilization was achieved at 515° F after 100 minutes. At 130 minutes the temperature began a gradual rise and restabilization and the test was stopped at 150 minutes (2-1/2 hours). During this run the torquemeter was not operating, but observation of the noise and vibration of the test stand confirmed the temperature indication of a smooth running bearing. Curves of race temperature versus time are given in figure 40.

A summary graph showing bearing, temperature, torque and horsepower versus engine speed for all Phase 2 (temperature-speed) testing is shown in figure . The drop in stabilization temperature from 8000 rpm to the higher speeds is attributed to the greater air circulation around the bearing housing caused by the increased drag of the turbine rotor disc. The temperature then climbed with an increase in speed. After this series of tests were completed, an additional run at 8000 rpm was made to verify the original temperatures. The results correlated very well, 540° F for initial test and 516° F for the check test.

Phase 3 testing consisted of running the unfired J-69 engine at 8000 rpm, with unheated carrier air and rear bearing, with thrust loading on the front bearing was varied from 0 to 500 pounds.

The original rear bearing housing, which also held the rear bearing heaters, was made of aluminum and had warped during Phase 2 testing due to the high temperatures of the bearing heaters preventing its use with the thrust loading system. A new housing and thrust piston of similar design but made of steel to prevent thermal distortion were made and installed in the engine. The new housing did not have provision for a heater to be placed at the rotor shaft axis, but as this axial heater only provided 3 percent of the heating power, its omission was not considered of major consequence.

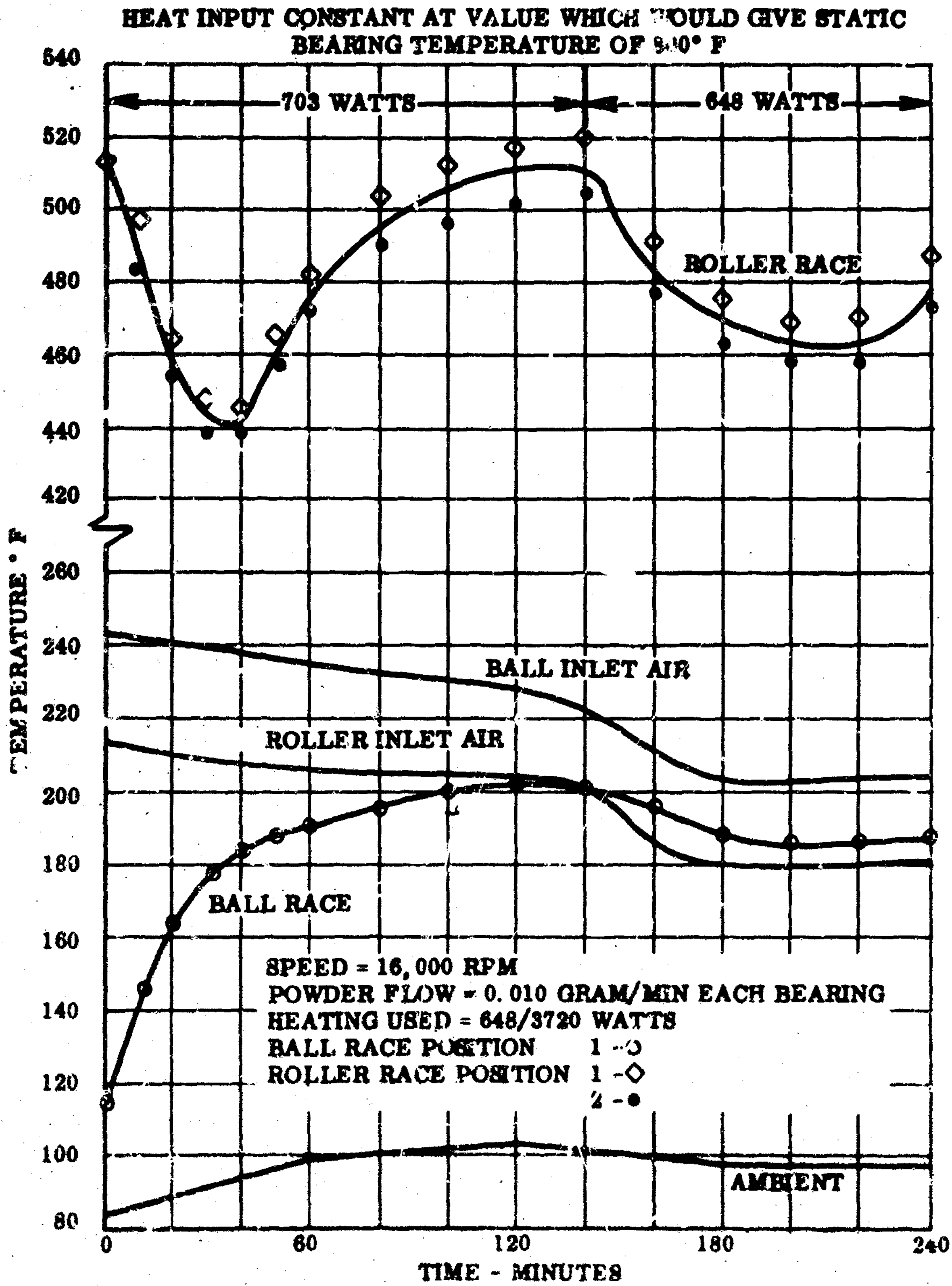

Figure 39. Engine Test E-40, Bearing Outer Race Temperature Versus Time

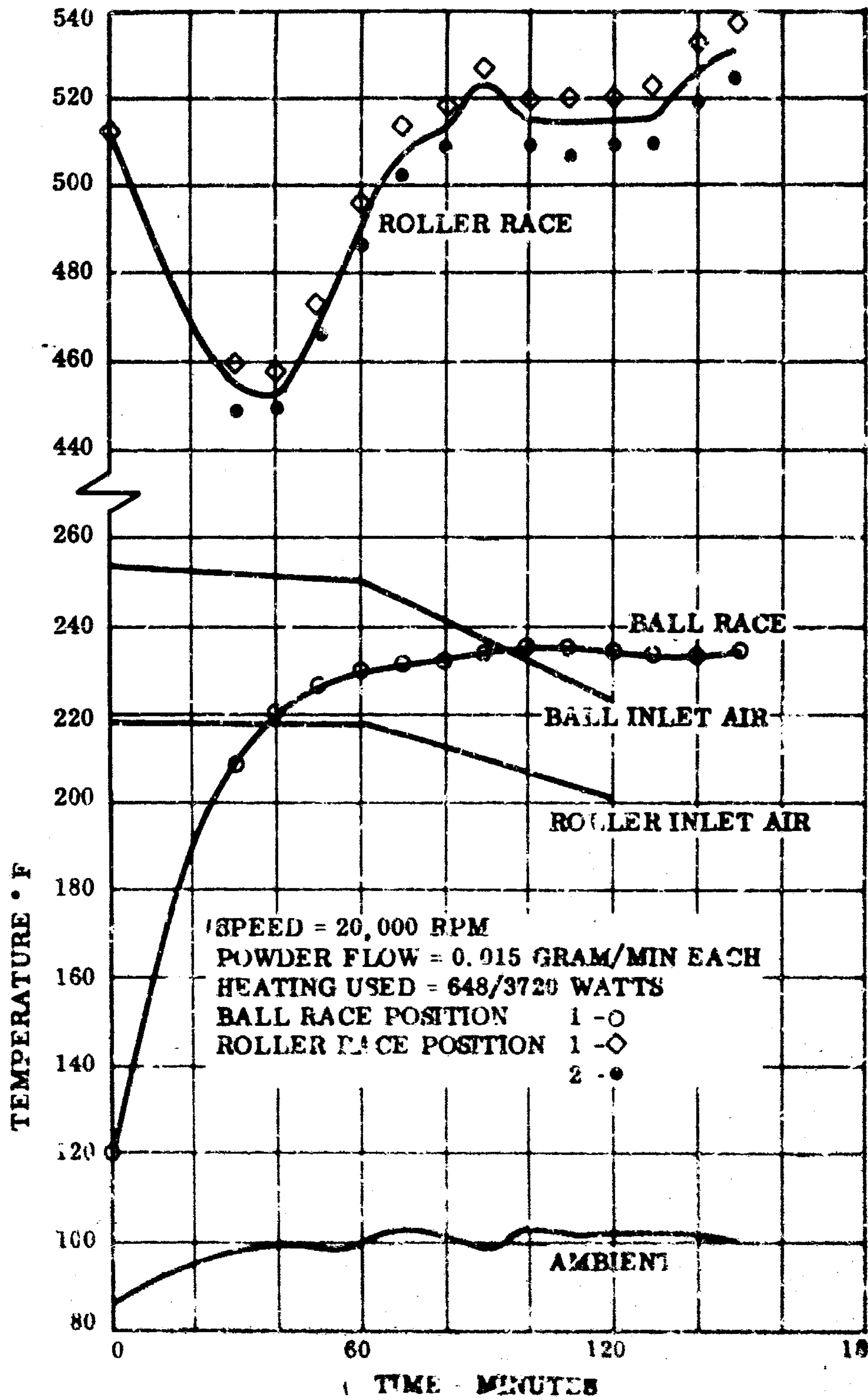

Figure 40. Engine Test E-41, Bearing Outer Race Temperature Versus Time

Test E-42 was performed as a checkout run of the new housing and heater arrangement. The results correlated well with those of a previous run made under similar conditions (E-33). Curves of race temperature versus time are given in figure 41.

Test E-43 was run with no thrust load until bearing temperatures had stabilized for one hour, at which time 70 pounds thrust load was applied through the rotor shaft to the front bearing. The front (ball) bearing temperature restabilized at about 20° F higher than with no load. The roller bearing restabilized at a lower temperature than the original, this being caused by thrust loading air leaking past the housing. Curves of temperature versus time are shown in figure 42.

Tests E-44 and E-45 were run successfully at 150 pounds and 250 pounds thrust load. Curves of race temperature versus time are shown in figure 43.

Test E-46 was run with 423 pounds thrust load because of excessive pressure drop in plumbing on the pneumatic lines supplying air to the thrust piston. The plumbing was reworked and test E-47 performed successfully at 500 pounds thrust load. Stabilization temperature at this load was 175° F, about 70° F higher than with no load. Curves of race temperature versus time for runs E-46 and E-47 are shown in figures 44 and 45.

Although the ball bearing performed well at all loads, there was about a 35 percent increase in power required to drive the engine as load was increased from 0 to 500 pounds. Curves of temperature versus load and total engine torque (ball and roller bearings) versus load are shown in figure 29.

Phase 4 testing (at varying speed, load, and temperature) was started at 8000 rpm with the bearings previously used through Phase 3 (B-5a, R-6a). Test E-48 was started at 8000 rpm with a heated rear bearing and 500 pounds thrust on the front bearing. Testing was started with the thrust air at room ambient temperature, but air leakage down the motor shaft had the effect of cooling the rear bearing, so thrust air temperature was raised to 500° F which gave rear bearing operating temperature of about 510° F. During this test the ball bearing outer race temperature did not stabilize and the test was shut down when this temperature reached 326° F. Engine teardown revealed good lubricant filming in both ball and roller bearings, but excessive wear and metal smearing was evident on the retainer guiding surfaces of the ball bearing. Before teardown, the bearings had accumulated 46 hours of testing at speeds from 8000 to 20,000 rpm. Curves of race temperature versus time for this run are shown in figure 46. Photographs of both bearings after teardown are presented in figures 47 through 50.

The engine was reassembled with new bearings of the same design as previously run, called B-5b, and R-6b, and test E-49 performed at 8000 rpm as a break in and calibration run of the new bearings. Performance correlated well with calibration of the previous set of bearings (test E-35). Curves of race temperature versus time are given in figure 51.

Test E-50 was performed at 8000 rpm with 500 pounds thrust load and was stopped after a successful run of 2 hours. Ball bearing temperatures were in good agreement with a similar run using the previous bearings (E-47). Curves of race temperature versus time are given in figure 52.

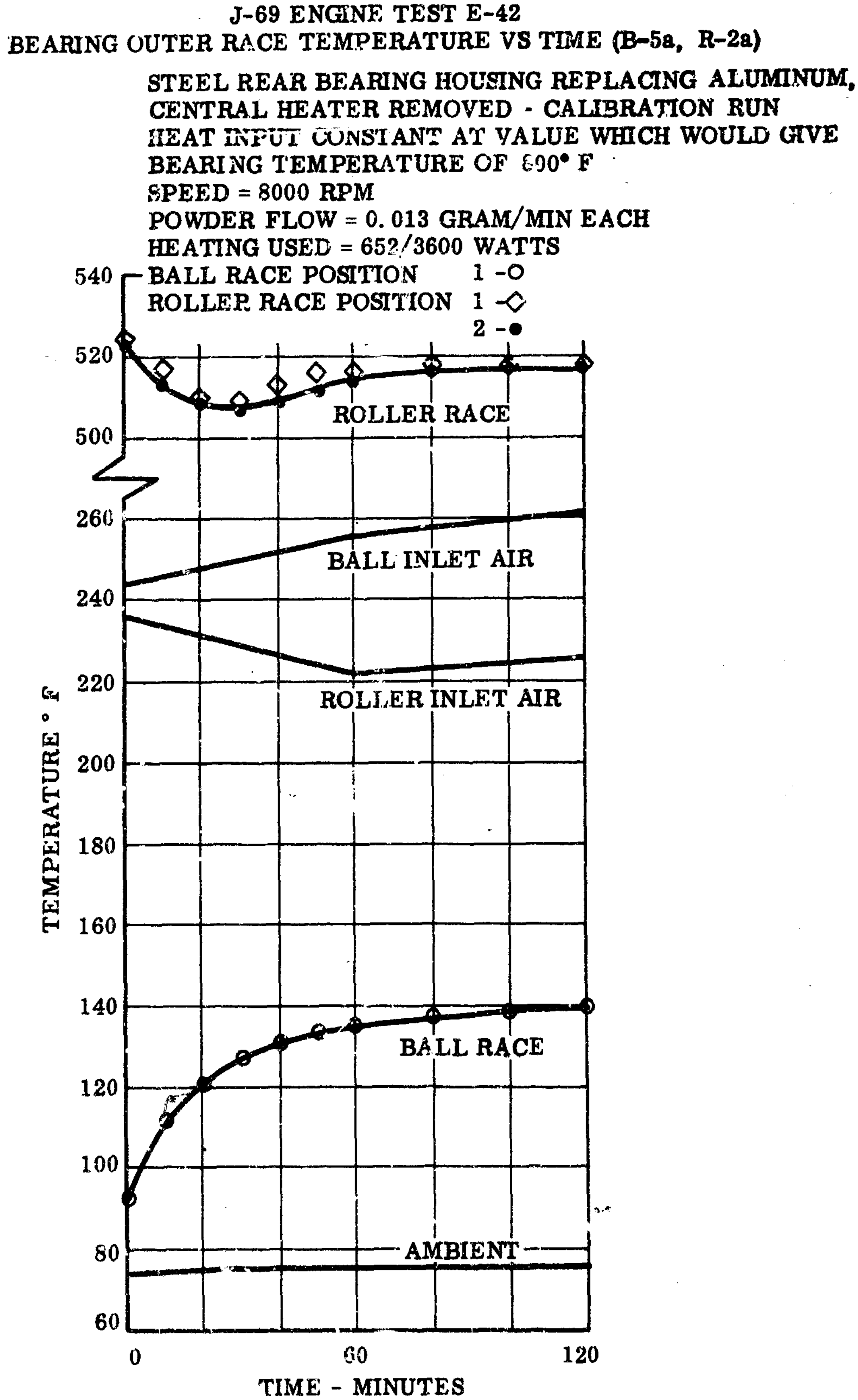

Figure 41. Engine Test E-42, Bearing Outer Race Temperature Versus Time

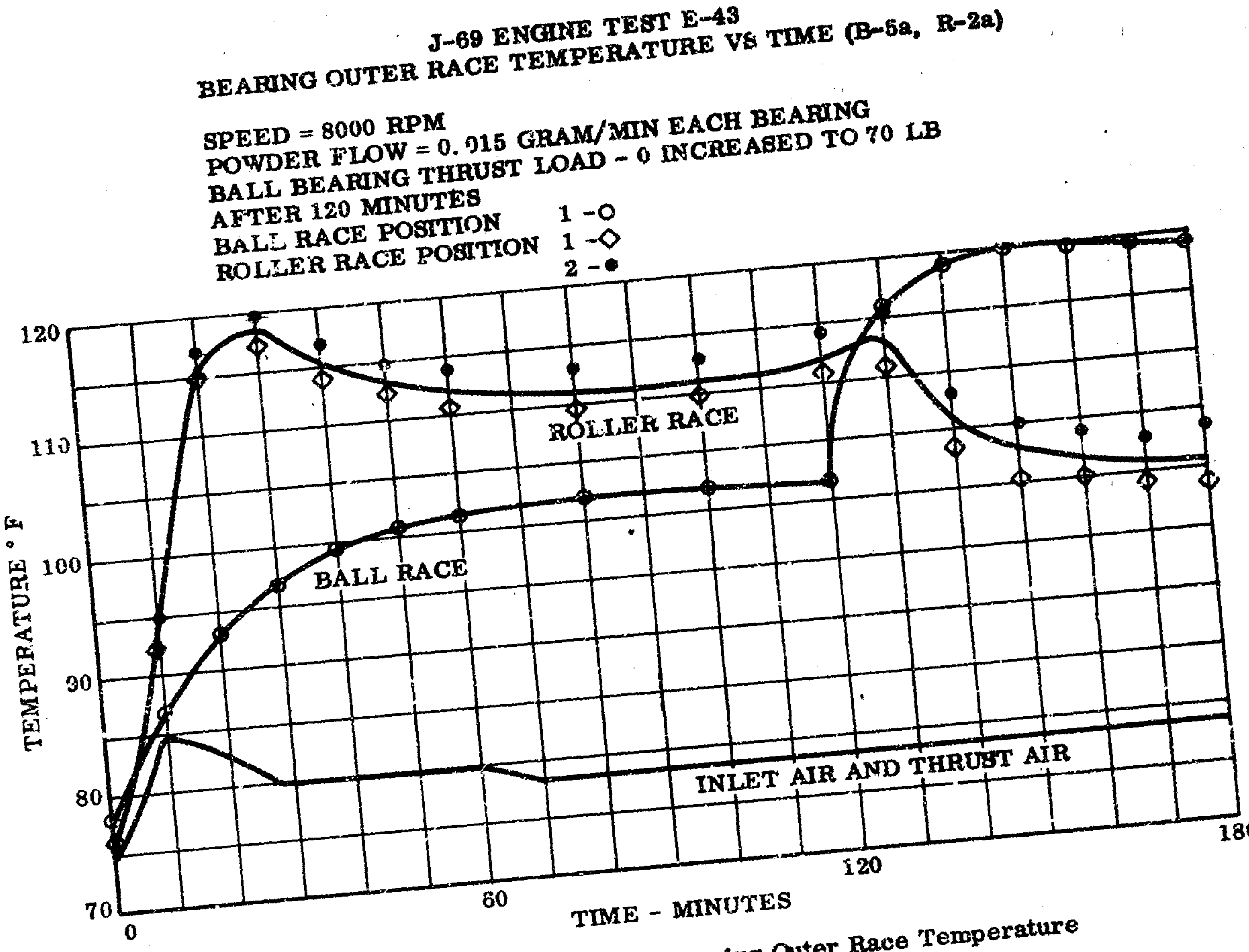

Figure 42. Engine Test E-43, Bearing Outer Race Temperature Versus Time

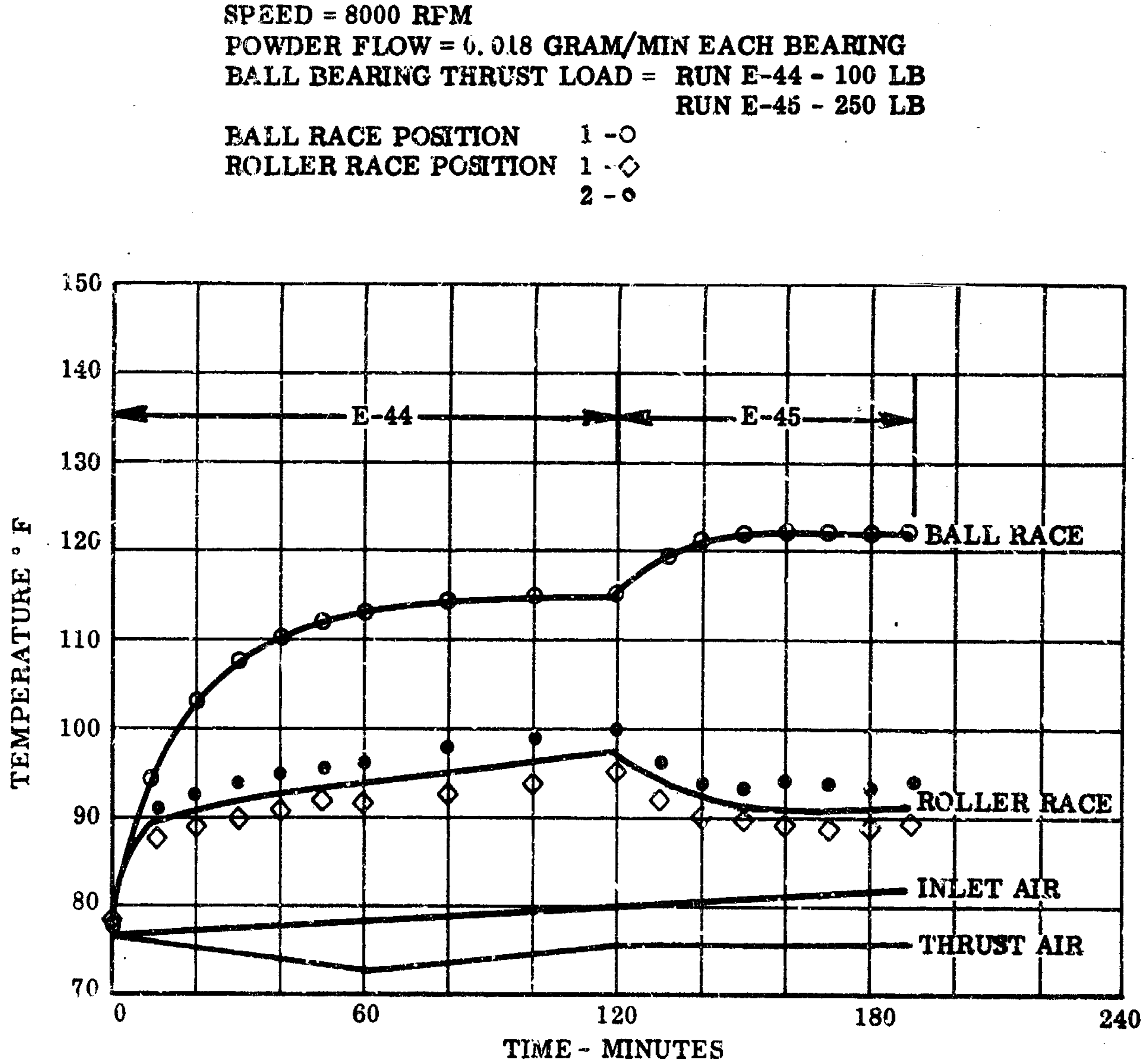

Figure 43. Engine Tests E-44 and E-45, Bearing Outer Race Temperature Versus Time

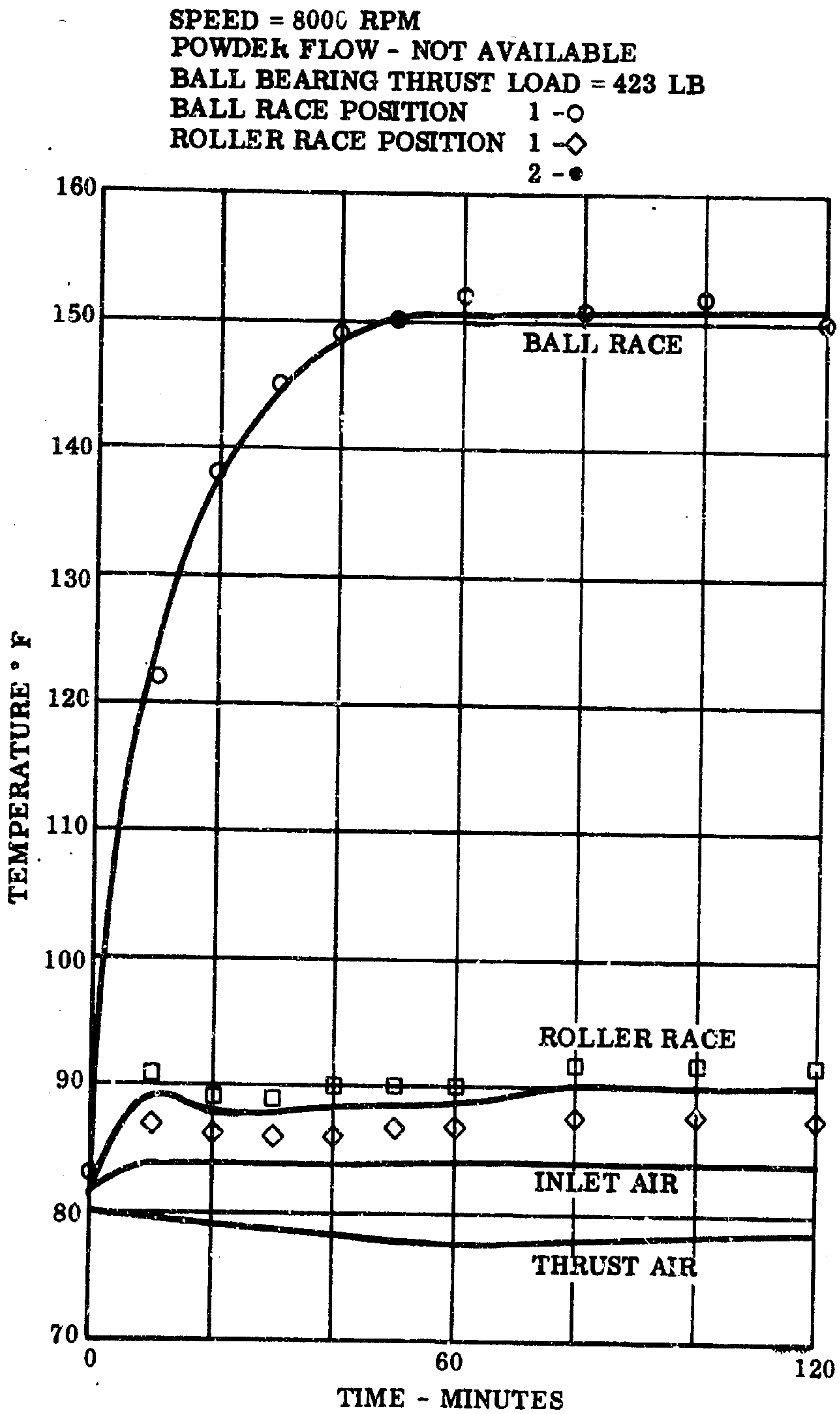

Figure 44. Engine Test E-46, Bearing Outer Race Temperature Versus Time

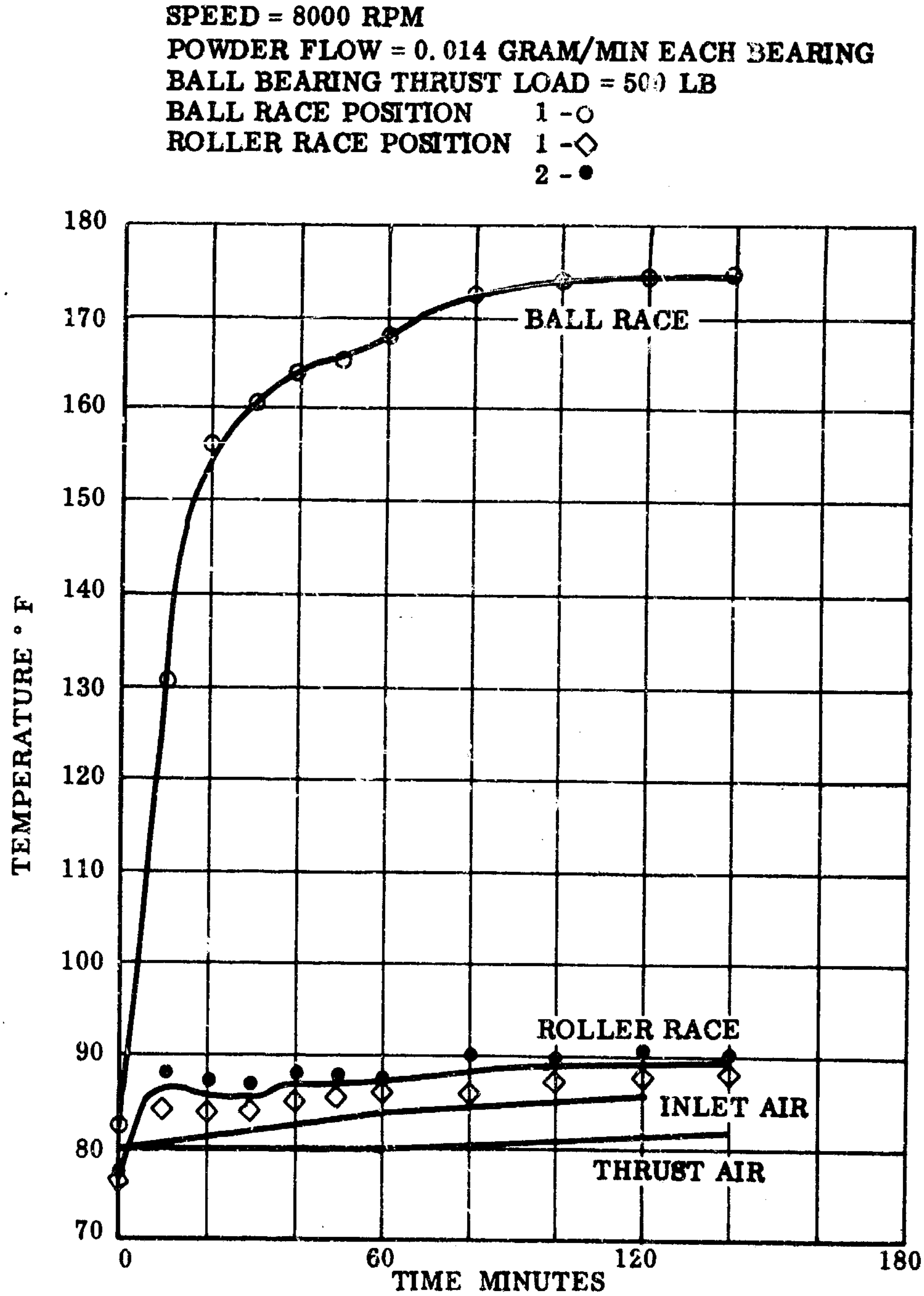

Figure 45. Engine Test E-47, Bearing Outer Race Temperature Versus Time

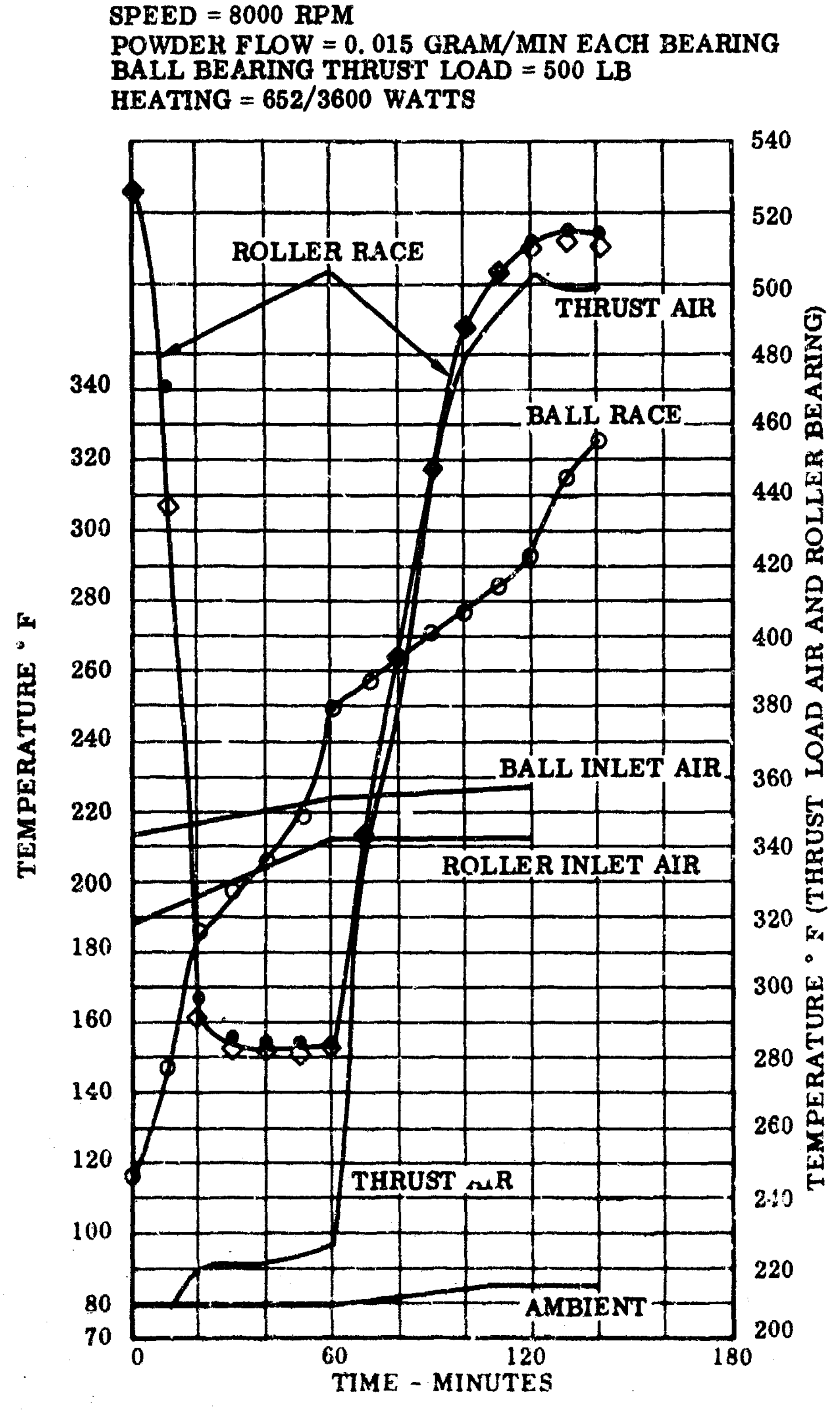

Figure 46. Engine Test E-48, Bearing Outer Race Temperature Versus Time

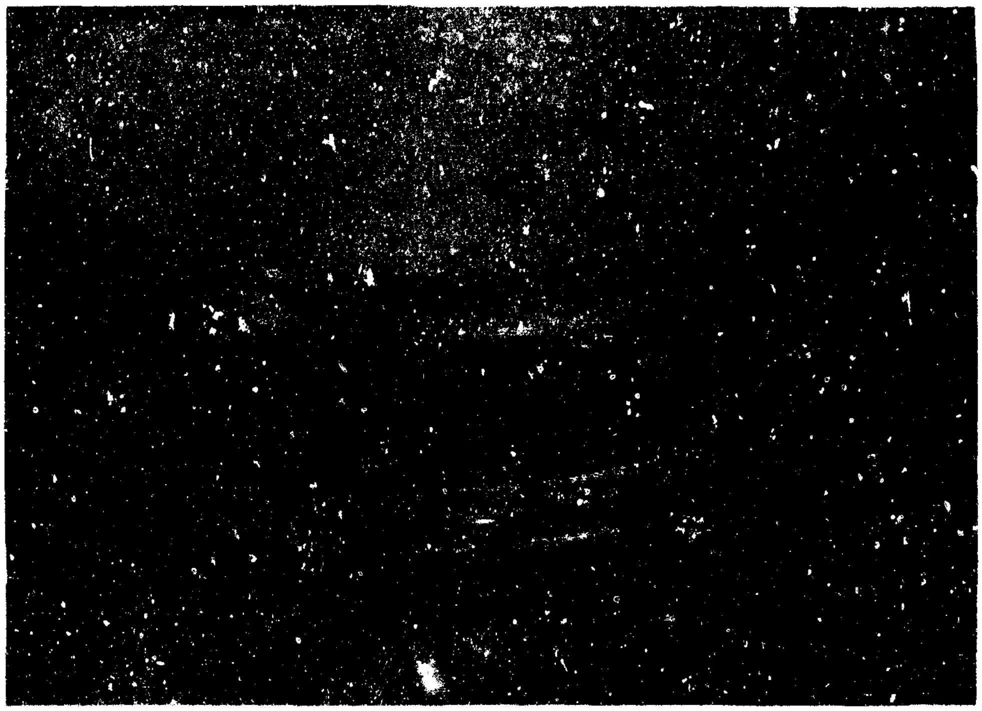

Figure 47. Test Bearing B-5a Outer Ring after Test E-48

Figure 48. Test Bearing B-5a Retainer after Test E-48

Figure 49. Test Bearing R-6a Outer Ring after Test E-48

Figure 50. Test Bearing R-6a Retainer and Rollers after Test E-48

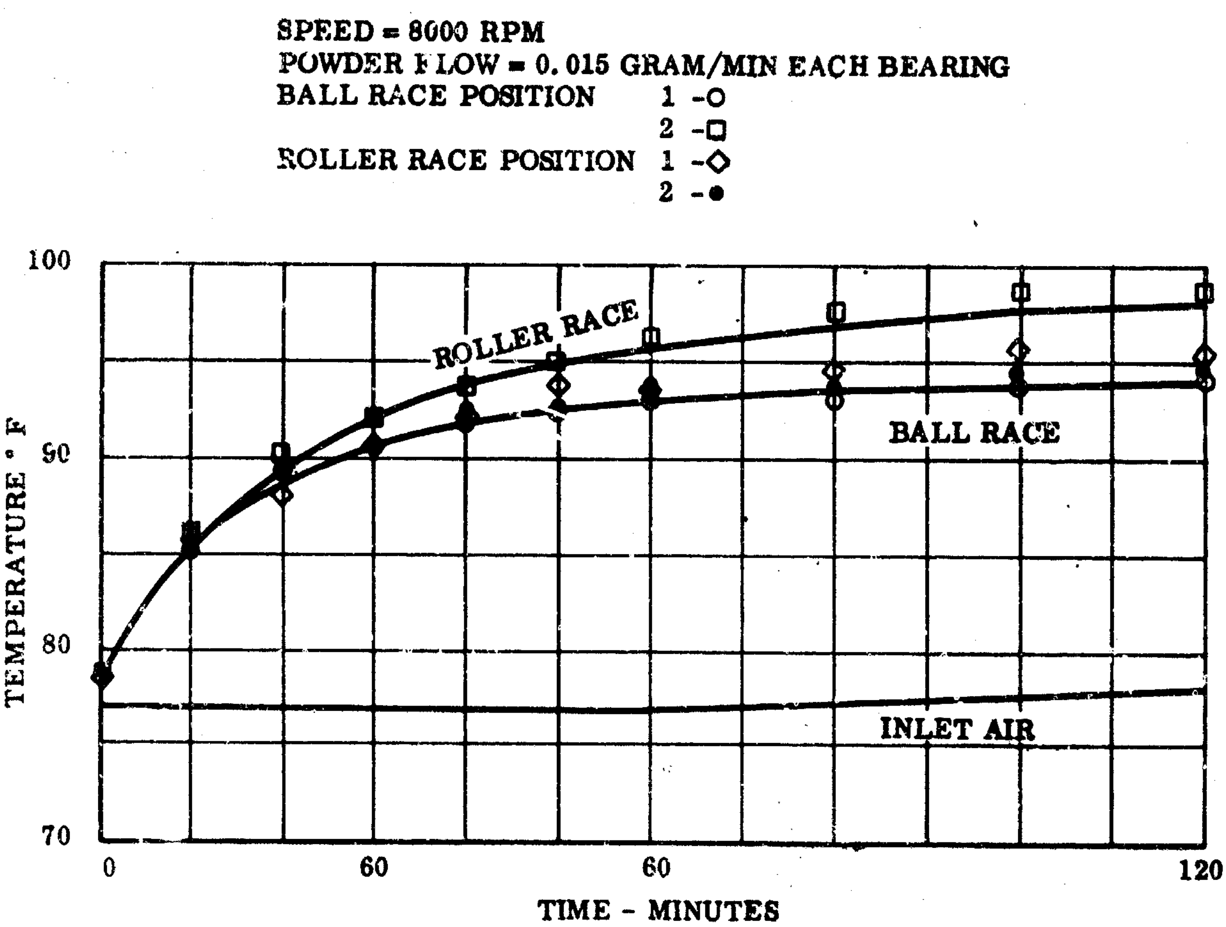

Figure 51. Engine Test E-49, Bearing Outer Race Temperature Versus Time

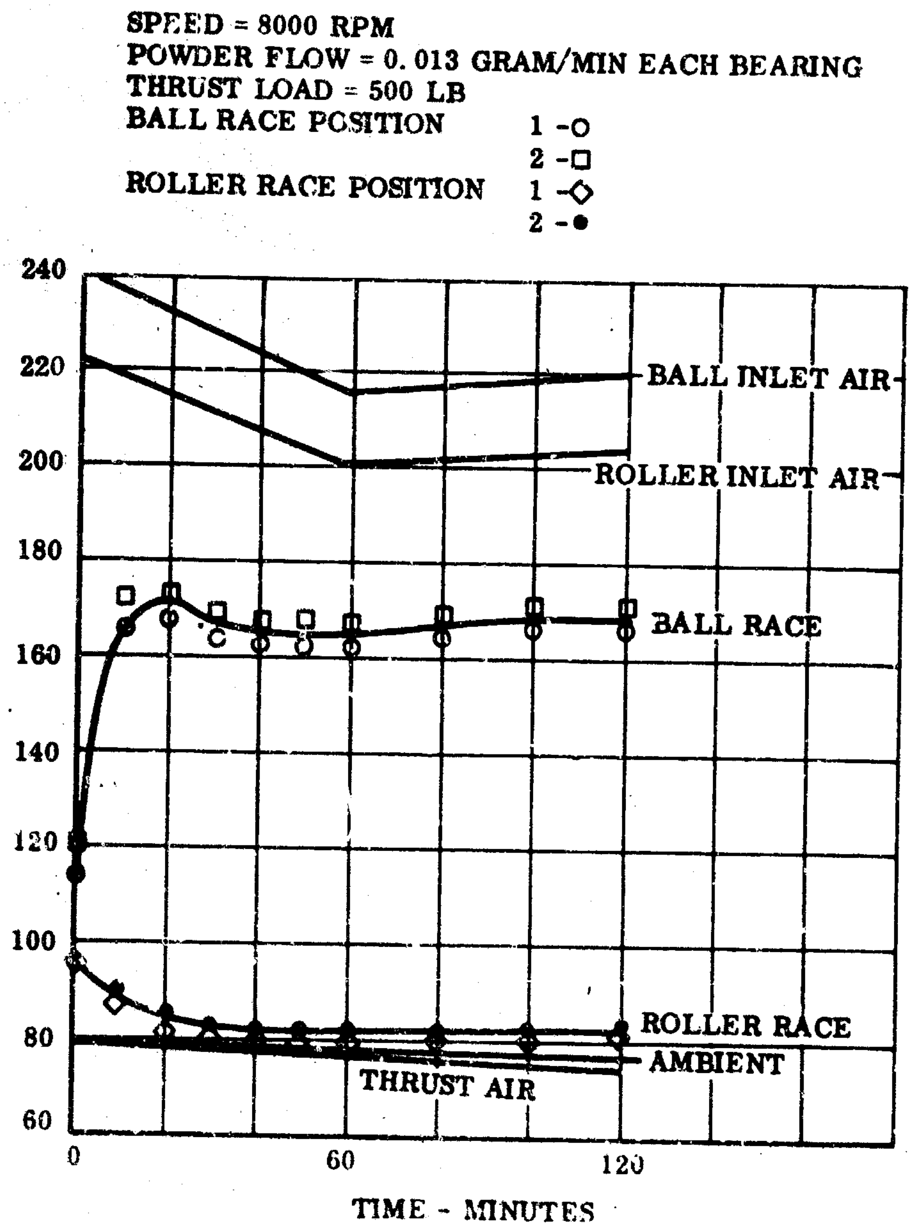

Figure 52. Engine Test E-60, Bearing Outer Race Temperature Versus Time

Test E-51 was started with 500 pounds thrust load at 12,000 rpm but was shut down when the ball bearing temperature ro to 350° F after 33 minutes of operation. Curves of race temperature versus time are shown in figure 53.

On teardown, it was seen that the two inner rings of the ball bearing had welded together at the inner race parting surface, this being accompanied by metal smearing and flaking as shown in figures 54 and 55. Severe wear and smearing was evident in the retainer ball pockets as shown in figure 56. Investigation showed that powder buildup or retainer flaking might have reduced internal bearing clearances and caused binding. This was confirmed by a failure analysis performed on the bearing by the manufacturer.

For test E-52, roller bearing R-6b was kept in the engine, but a new ball bearing of previously untested design (B-4) was installed as the front bearing. The balls and rings of this bearing are of M-50 steel and the retainer of Monel S. The retainer has an X-shaped cross section designed theoretically to provide for better lubricant flow through the bearing. Test E-52 was a bearing break in and calibration run and was stopped after successful 2 hour run at 8000 rpm. Curves of race temperature versus time are shown in figure 57.

Test E-53 was run for 2.3 hours, with both bearings achieving stabilized temperature operation at 8000 rpm with 50 pounds thrust on the front bearing. Ball bearing stabilization temperature under this load was about 35° F higher than for test E-52 with no thrust load, which was caused in part by the use of heated lube carrier air for test E-53. Curves of race temperature versus time are shown in figure 58.

Test E-54 was started at 8000 rpm with 100 pounds thrust on the front bearing. The test started well and the ball bearing was approaching stabilized operation after 50 minutes, when the ball bearing temperature and engine torque suddenly increased, requiring test shutdown. Inspection revealed a good lubricant film on the ball bearing races and balls, but very poor filming on the Monel retainer rubbing surfaces. On these surfaces the lubricant appeared to have caked unevenly and smeared, producing an uneven sliding surface. It was decided to discontinue testing of this retainer design in favor of the more successful design (B-5 series) which had run at 8000 rpm with up to 500 pounds thrust load. Curves of race temperature versus time for test E-54 are shown in figure 59 and a photograph of the retainer rubbing surfaces in figure 60.

For test E-55, roller bearing R-6b was kept in the engine, but a new ball bearing (B-5c) of the type previously tested to 20,000 rpm with no load, was installed. Run E-55 was a checkout and calibration run at 8000 rpm with no load. There was a rapid rise of roller bearing temperature soon after the run was started, probably caused by some powder clogging, but bearing temperature thereafter dropped off and stabilized with no further problems. Stabilization temperatures for this run were about 10° F higher than the previous comparable run (E-49) but this was considered only a minor discrepancy, as bearing performance was otherwise very smooth. Curves of race temperature versus time are shown in figure 61.

Test E-56 was run at 8000 rpm with heated carrier air and 50 pounds thrust load on the ball bearing. Ball bearing temperature showed a very slight (5° F) temperature variation during the test, but the bearing was considered to have achieved stabilized operation, and the test was stopped after 140 minutes. Ball bearing operating temperature was about 50° F higher than in run E-55, but most of this increase was attributable to the heated carrier air. There was no corresponding increase in

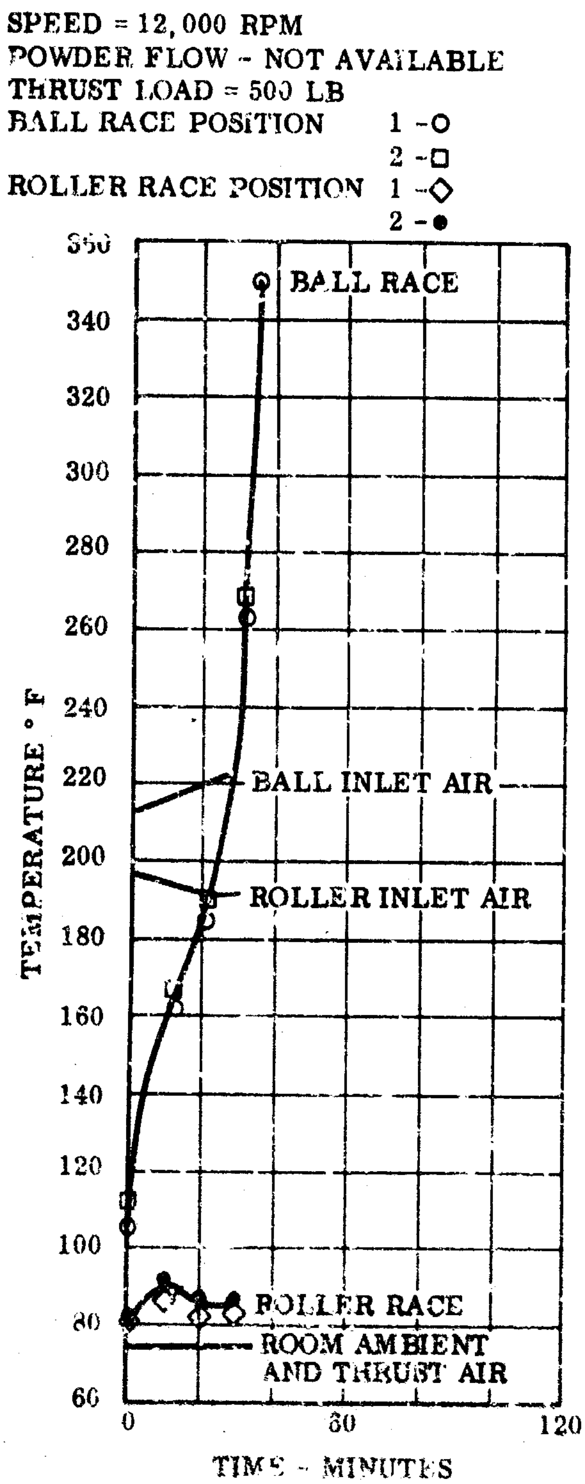

Figure 53. Engine Test F-51, Bearing Outer Race Temperature Versus Time

Figure 54. Test Bearing B-5b Inner Rings after Test E-51

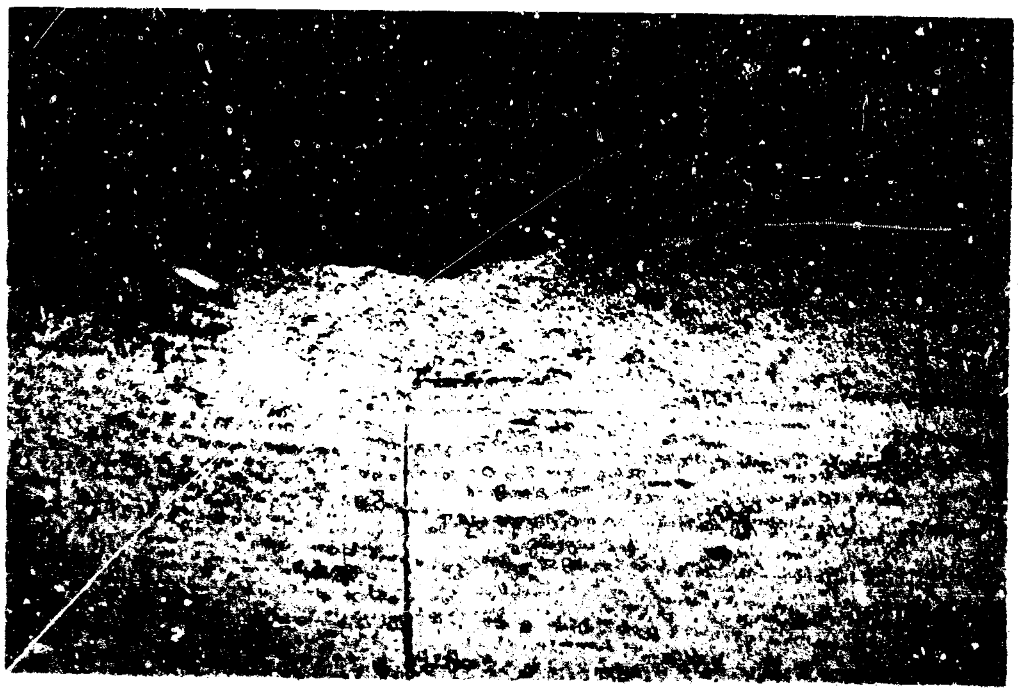

Figure 55. Test Bearing B-5b Inner Ring Cross Section Showing Weld at Race Surface

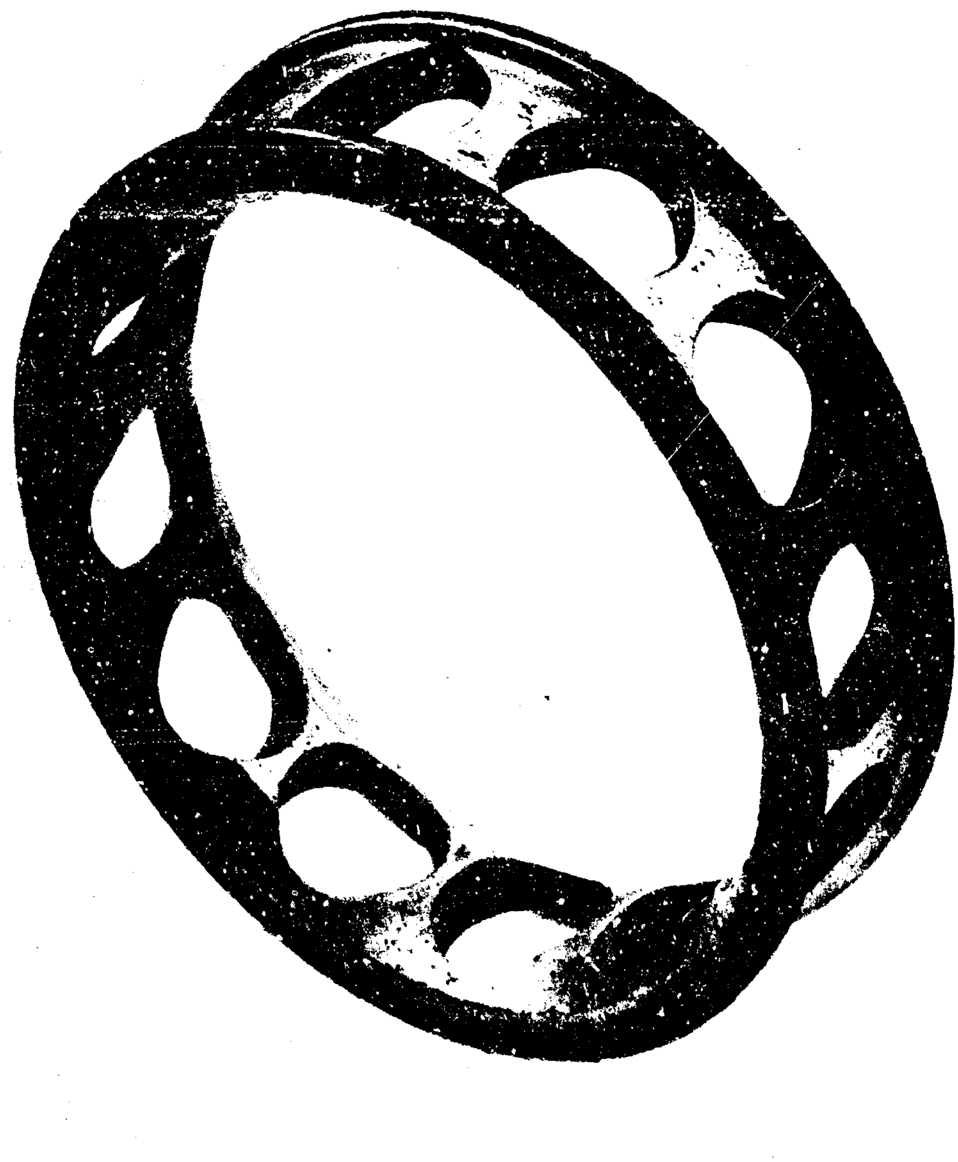

Figure 56. Test Bearing B-5b Retainer after Test

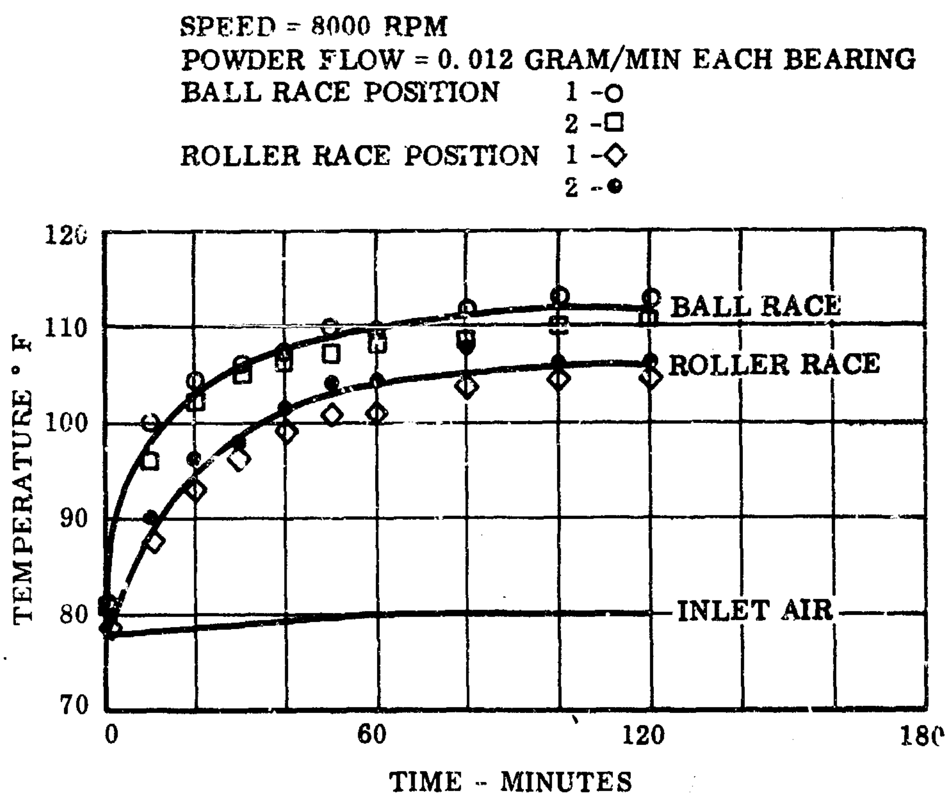

Figure 57. Engine Test E-52, Bearing, Outer Race Temperature Versus Time

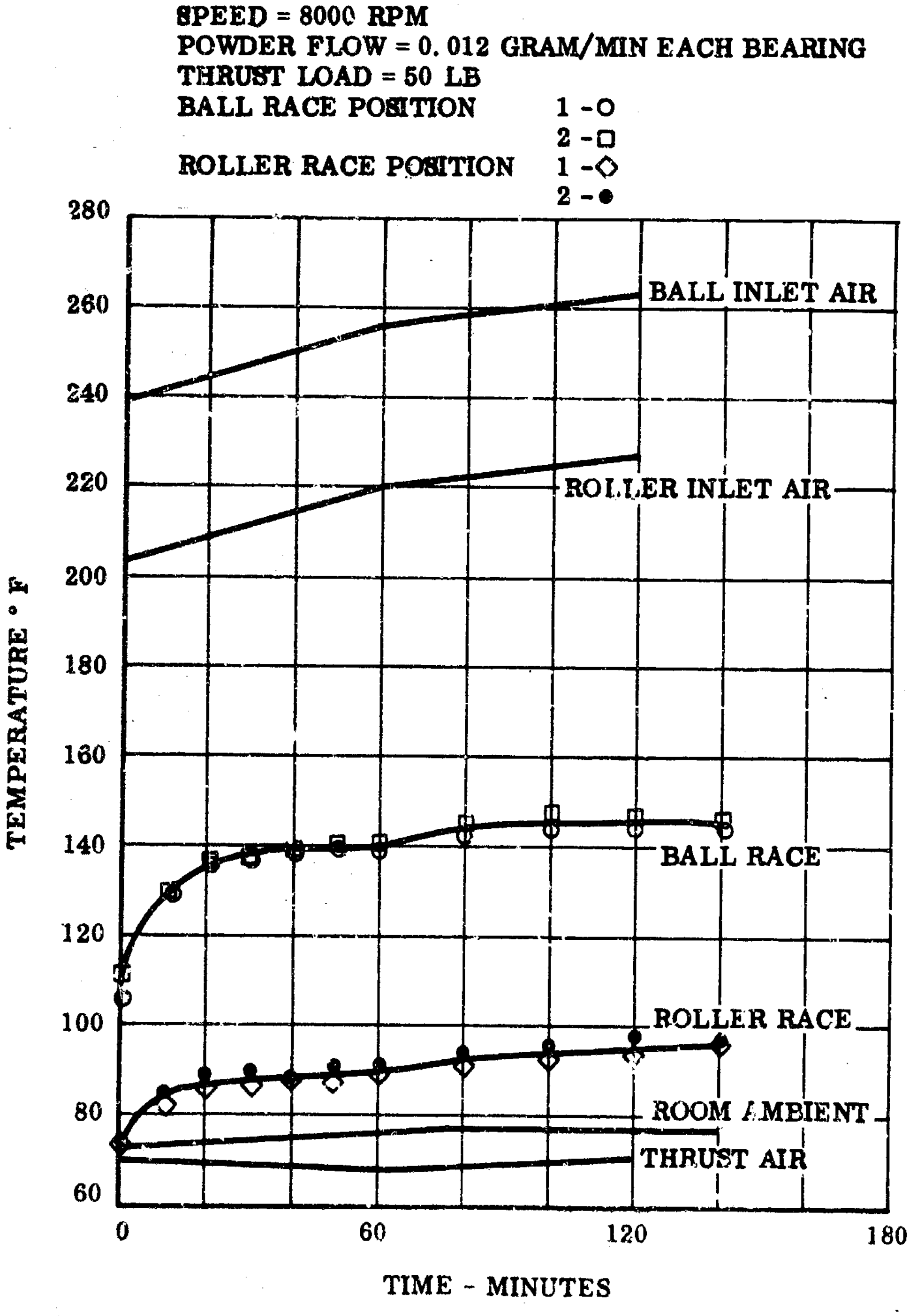

Figure 58. Engine Test E-53, Bearing Outer Race Temperature Versus Time

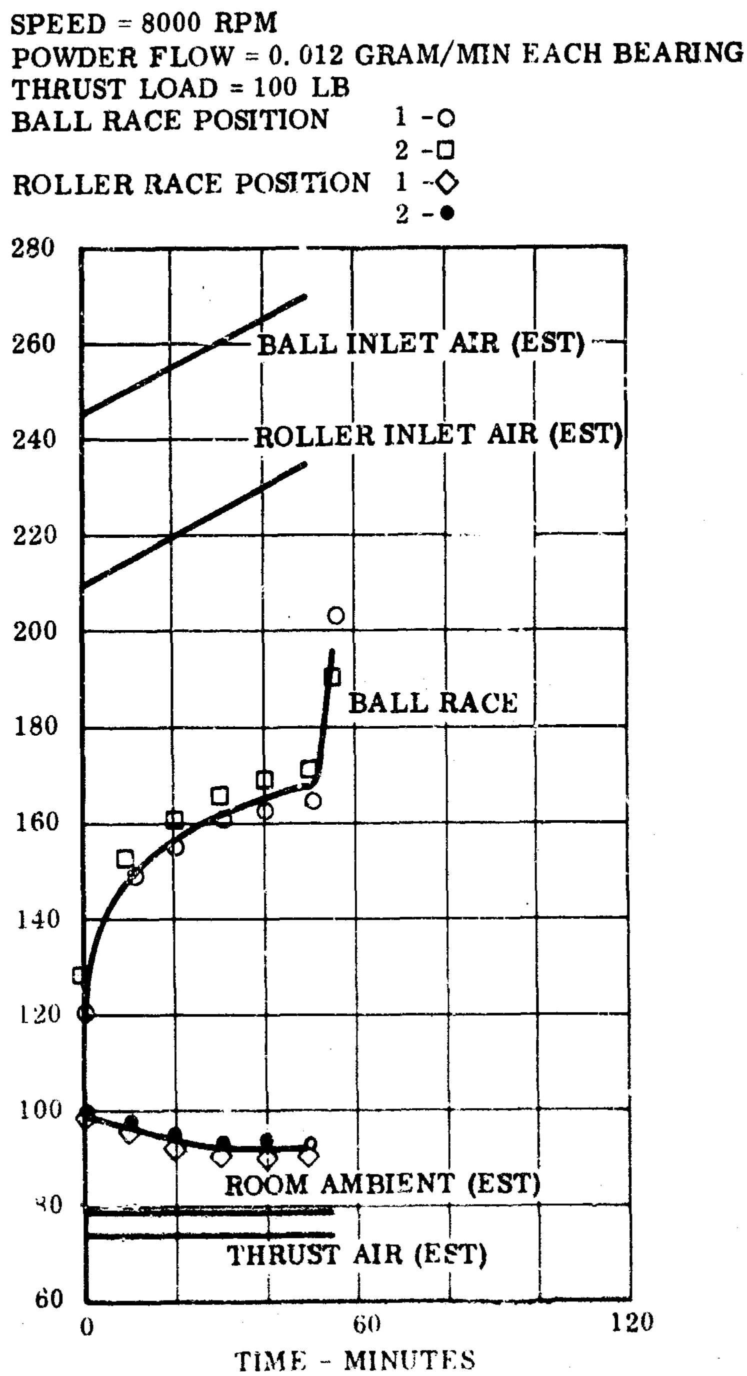

Figure 59. Engine Test E-54, Bearing Outer Race Temperature Versus Time

Figure 60. Test Bearing B-4 After Test E-54

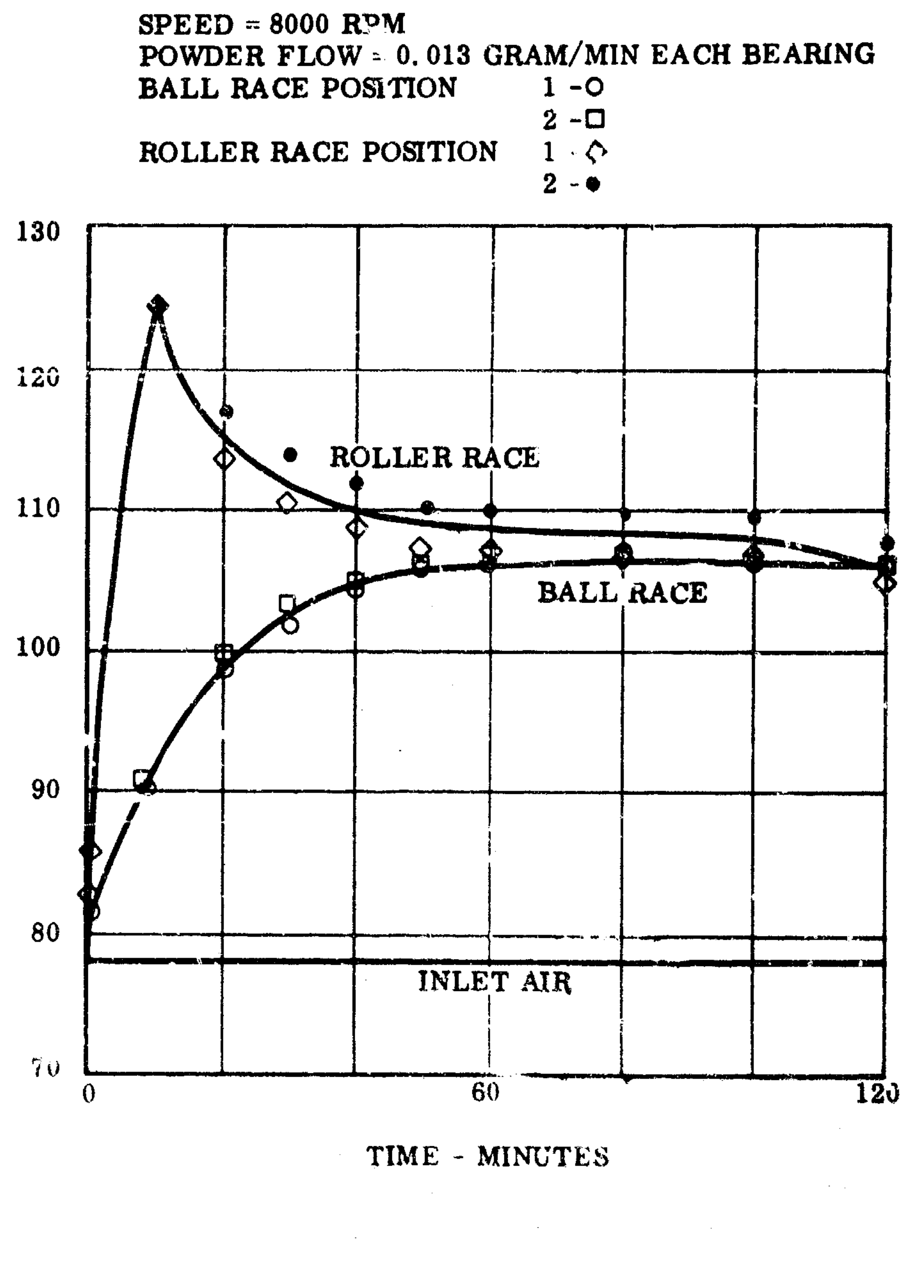

Figure 61. Engine Test E-55, Bearing Outer Race Temperature Versus Time

roller bearing temperature from the previous test, but this was due to the cooling effect of leaking thrust air. Curves of race temperature versus time are shown in figure 62.

Test E-57 was run with 100 pounds thrust load at 8000 rpm, and stabilized operation of the ball bearing was obtained at about the same temperature as with 50 pounds load. Curves of race temperature versus time are shown in figure 63.

Testing at 12,000 rpm with 50 pounds thrust load was performed with test run E-58. With heated carrier air, the ball bearing reached a stabilized operating temperature of 165° F and remained at this level until the test was stopped after two hours. This was an increase of about 10° F over the 8000 rpm operating condition. Curves of race temperature versus time are shown in figure 64.

Test E-59 was performed as a checkout run with a new rear labyrinth and rear bearing. The run was successfully completed at 8000 rpm with no thrust load and with lubricant carrier air at room ambient temperatures. Curves of race temperature versus time are shown in figure 65.

Tests E-60 and E-61 were repeats of previous runs at 8000 and 12,000 rpm respectively with heated carrier air and 50 pounds thrust load. Time-temperature curves for these runs are shown in figures 66 and 67.

Test E-62 was successfully performed at 16,000 rpm with 50 pounds thrust and heated carrier air. During this run the ball bearing temperature stabilized at about 195° F, as shown in figure 68, which was about 50° F greater than at 8000 rpm. Immediately following test E-62, the engine speed was being increased for a run at 20,000 rpm when, at about 19,000 rpm, an aluminum shroud over the turbine wheel broke, inducing severe vibrations in the engine and causing an immediate shutdown. Upon teardown, there was seen to be no damage to any of the rotating parts of the engine, and it was decided to attempt continued testing with the same bearings and no shroud over the turbine disc. The engine rotor assembly was dynamically balanced with the unshrouded turbine disc, and test E-63 was performed as a checkout at 8000 rpm with no load. As shown in figure 69, both bearings achieved temperature stabilization at a low level, indicating that there had been no damage incurred to the bearings when the shroud broke. The torque, however, was about three times as high as previous similar runs with a shrouded turbine disc, indicating that windage losses caused by the exposed fir tree footings would be in excess of the drive system capabilities at high speeds (16,000 and 20,000 rpm).

In lieu of installing a new turbine shroud, a simulated turbine disc with no fir tree footings was fabricated and used in place of the original turbine disc on the engine rotor shaft. Test E-64 was performed as a checkout run of the new engine assembly at 8000 rpm, during which the bearings performed satisfactorily as shown in figure 70. Immediately following this run, a high speed checkout run of 4-1/2 hour duration was made. This included one hour's running at 16,000 with 50 pounds thrust load, and several minutes running at 20,000 rpm with no load. Torque readings and engine noise indicated that the bearings were performing well throughout this run, but accurate temperature readings could not be obtained because of difficulties encountered in the temperature readout instrumentation.

Test E-65 was run at 16,000 rpm as a comparison with run E-62. During this test, thrust load on the front bearing was varied from 0 to 50 pounds. With no thrust

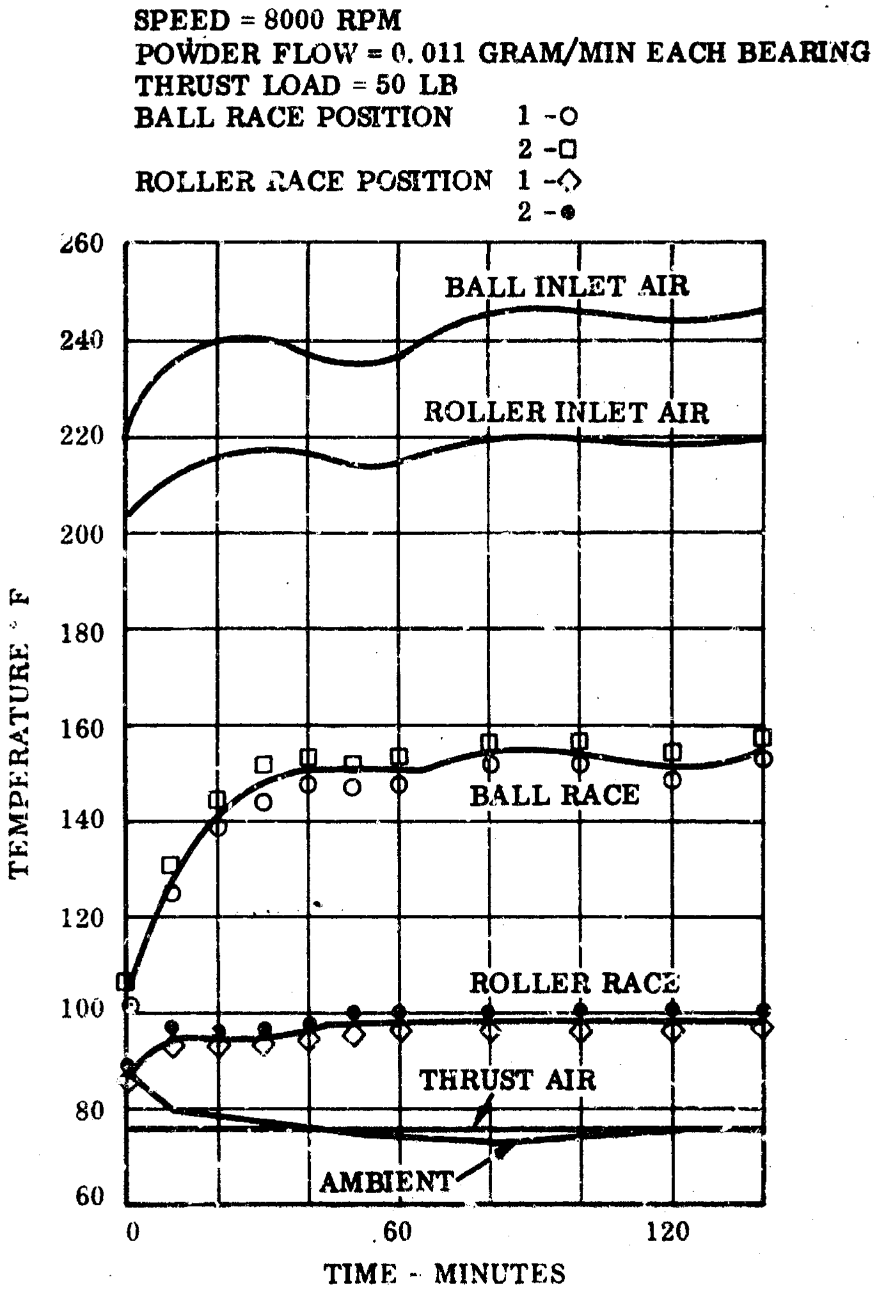

Figure 62. Engine Test E-56, Bearing Outer Race Temperature Versus Time

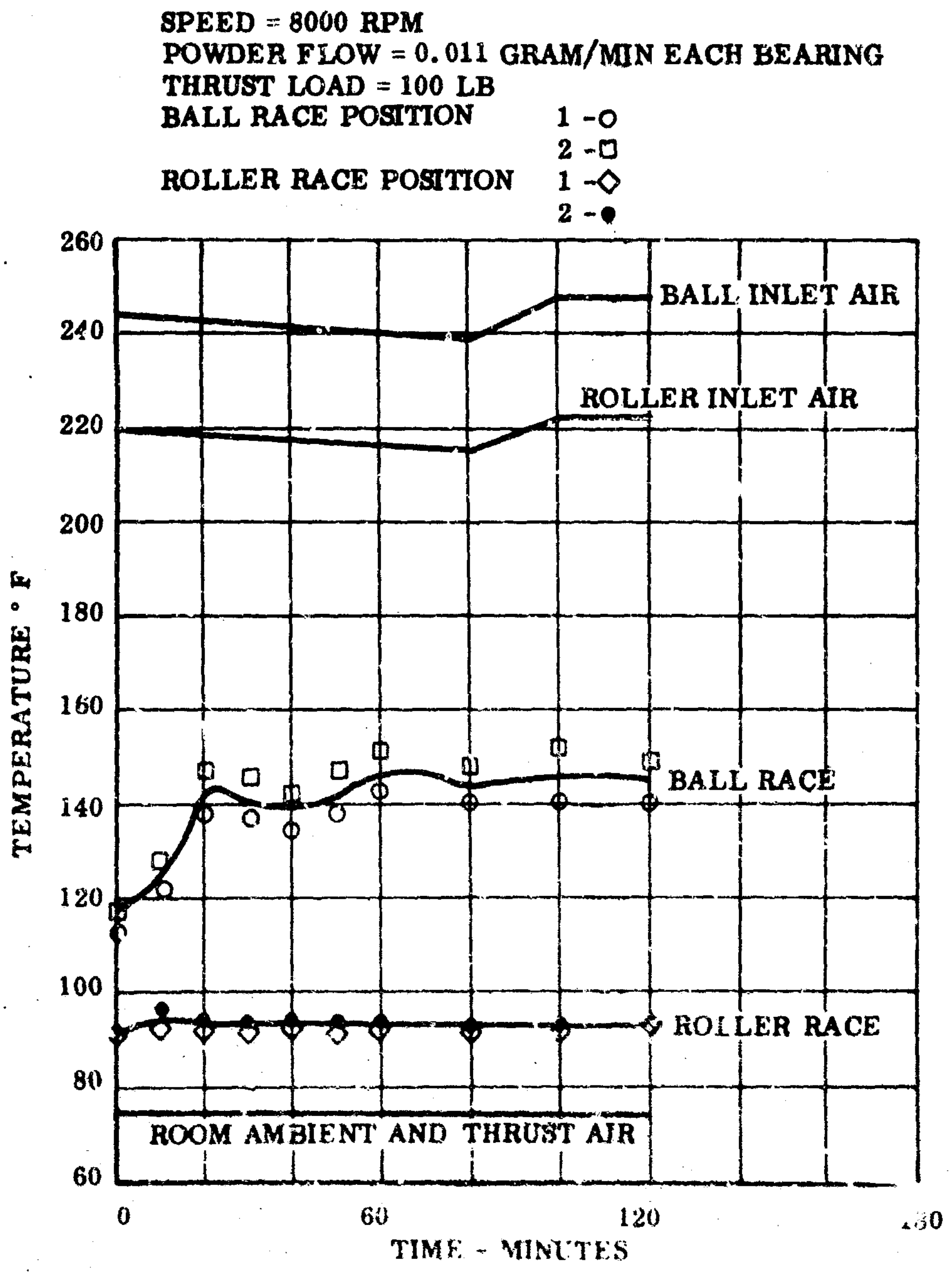

Figure 63. Engine Test E-57, Bearing Outer Race Temperature Versus Time

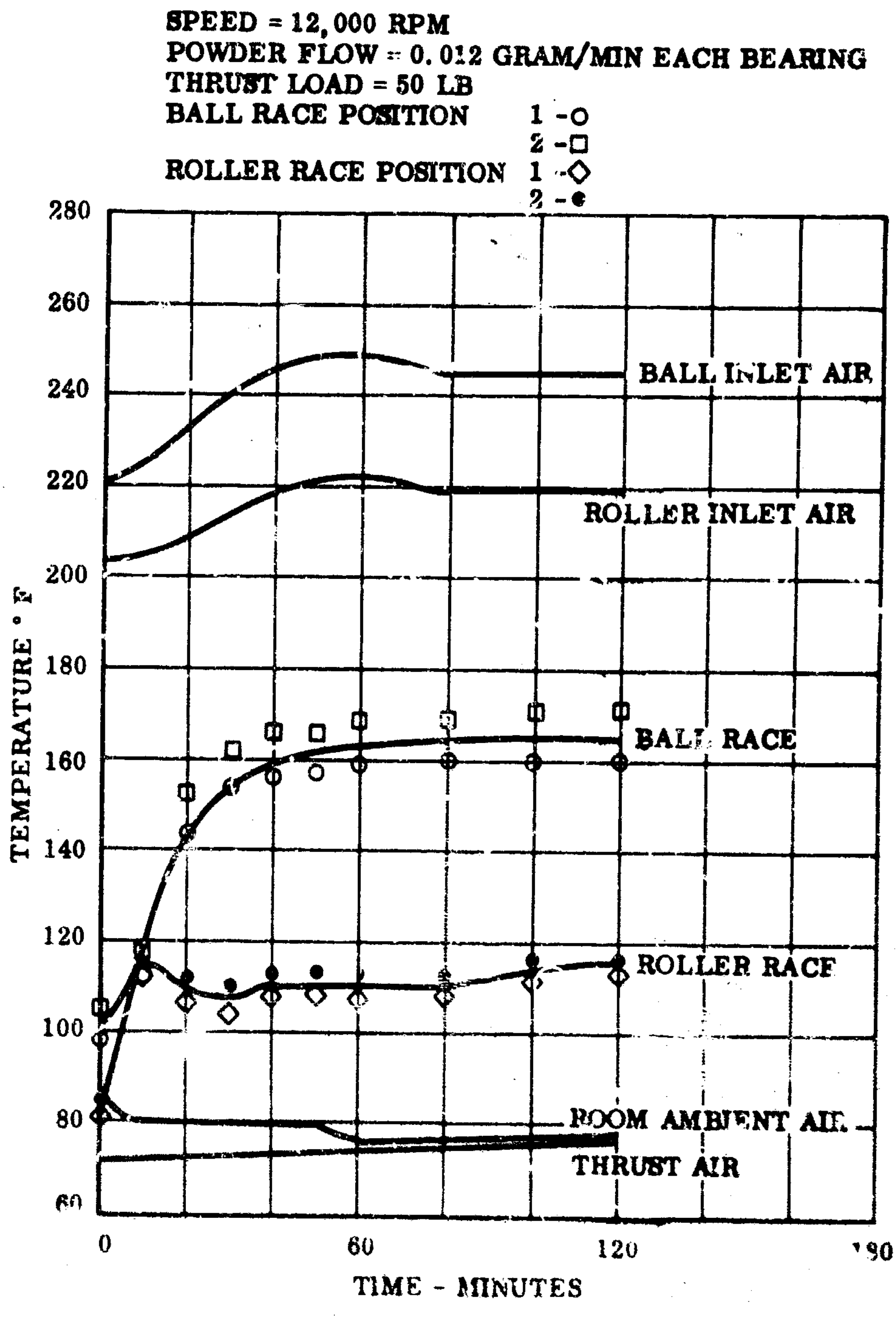

Figure 64. Engine Test E-58, Bearing Outer Race Temperature Versus Time

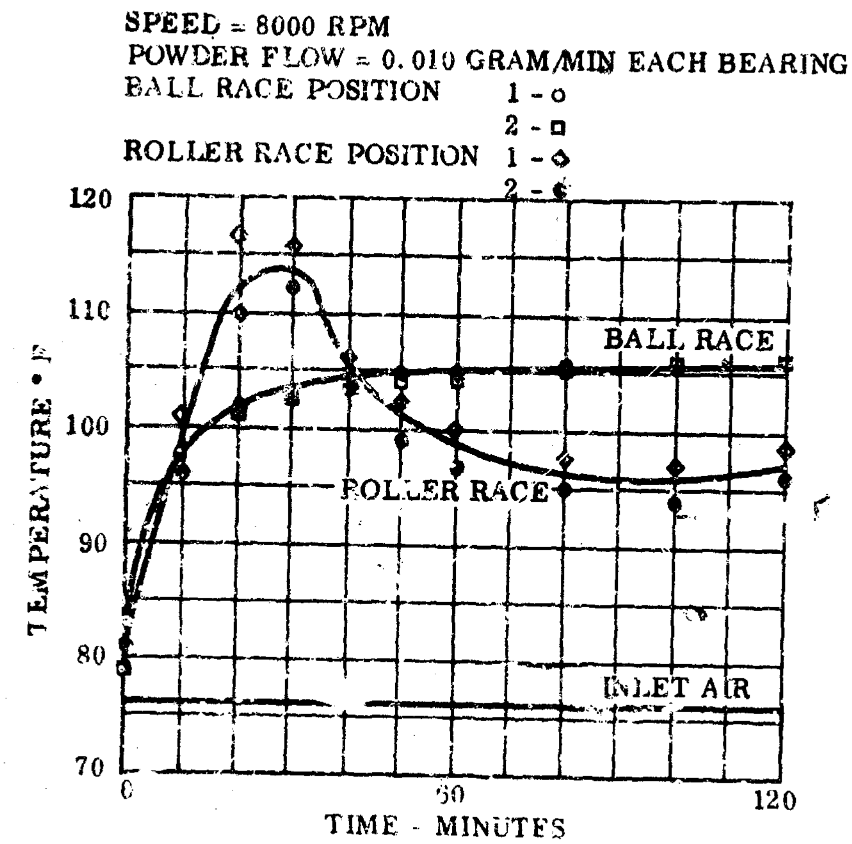

Figure 65. Engine Test E-59, Bearing Outer Race Temperature Versus Time

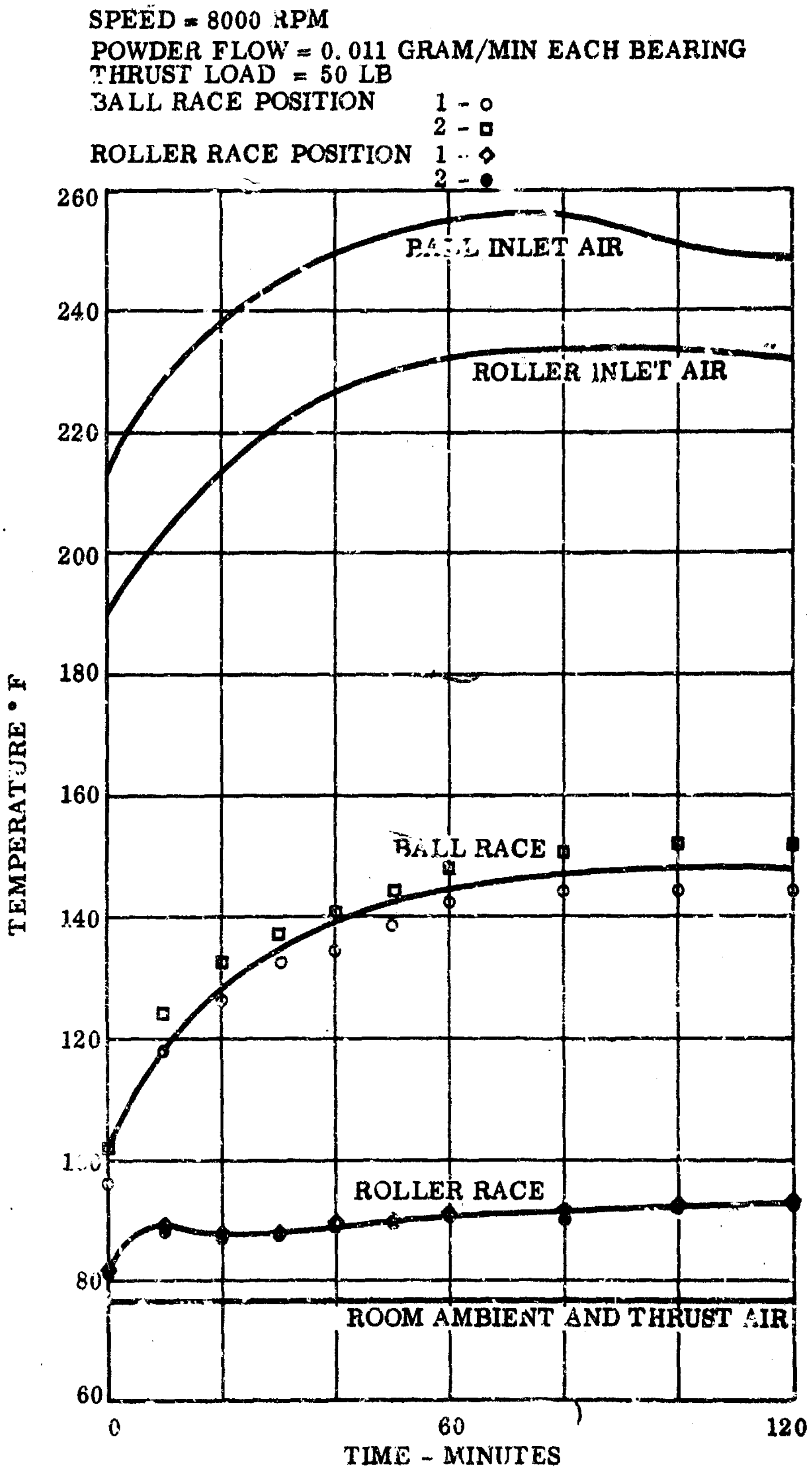

Figure 66. Engine Test E-60, Bearing Outer Race Temperature Versus Time

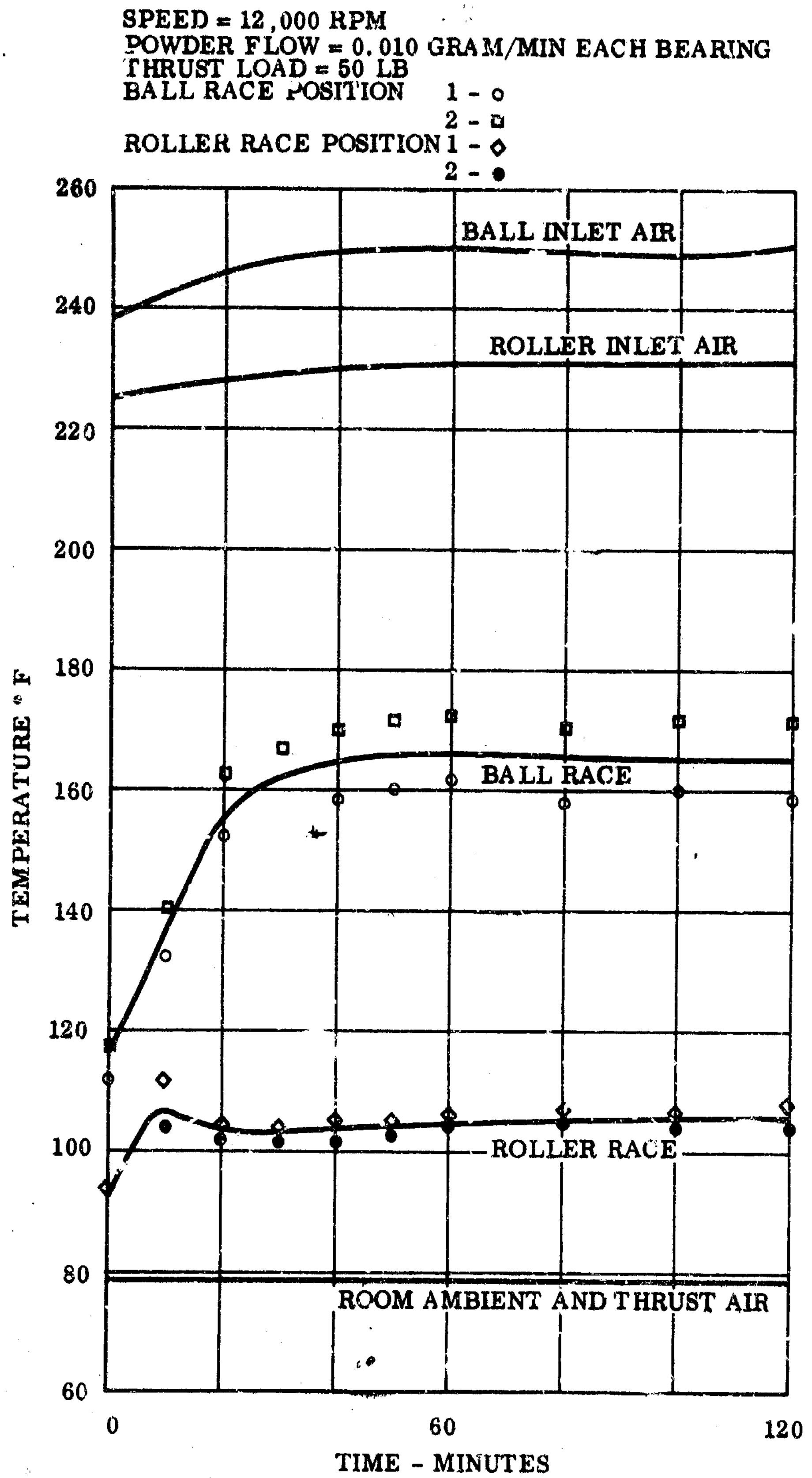

Figure 67. Engine Test E-61, Bearing Outer Race Temperature Versus Time

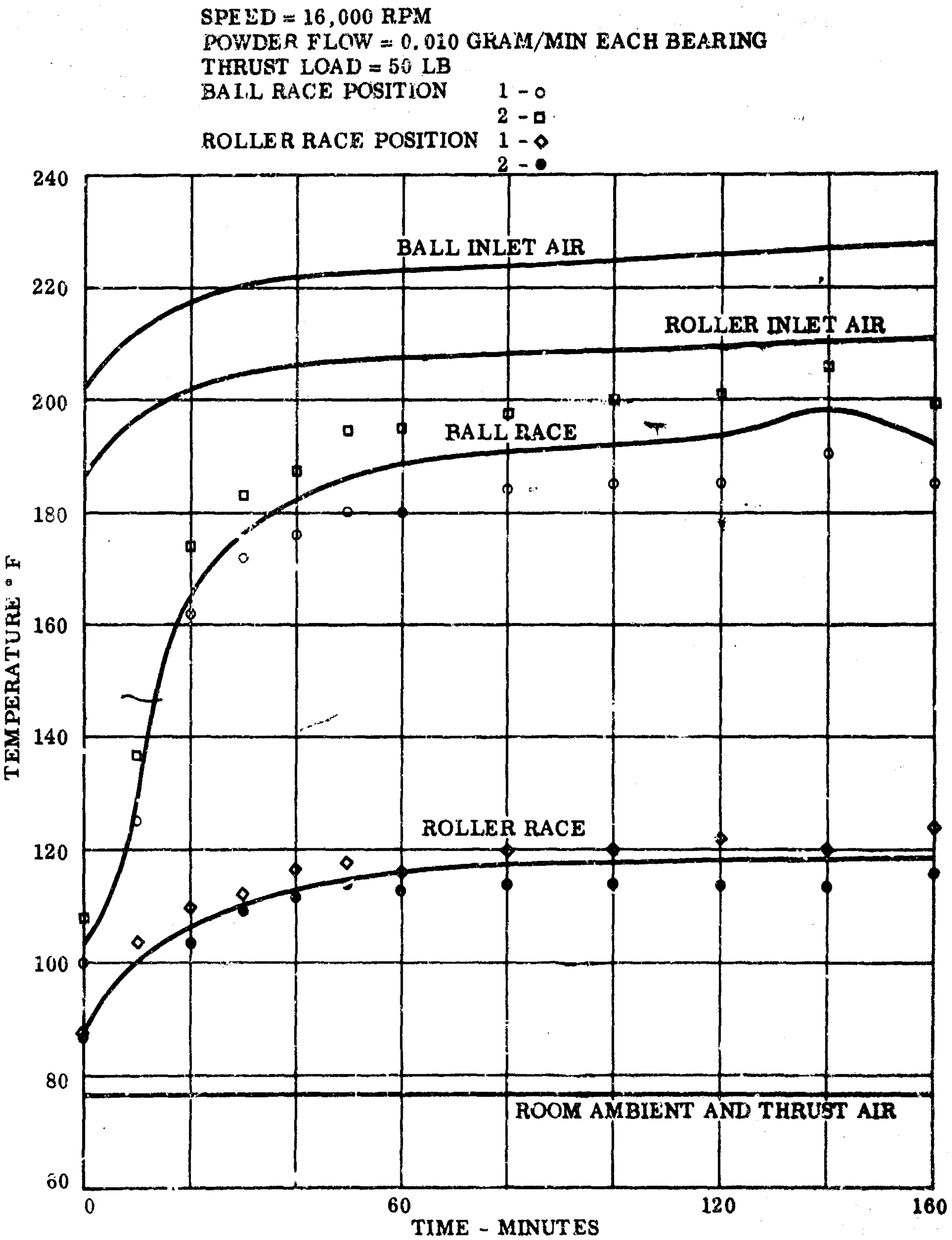

Figure 68. Engine Test E-62, Bearing Outer Race Temperature Versus Time

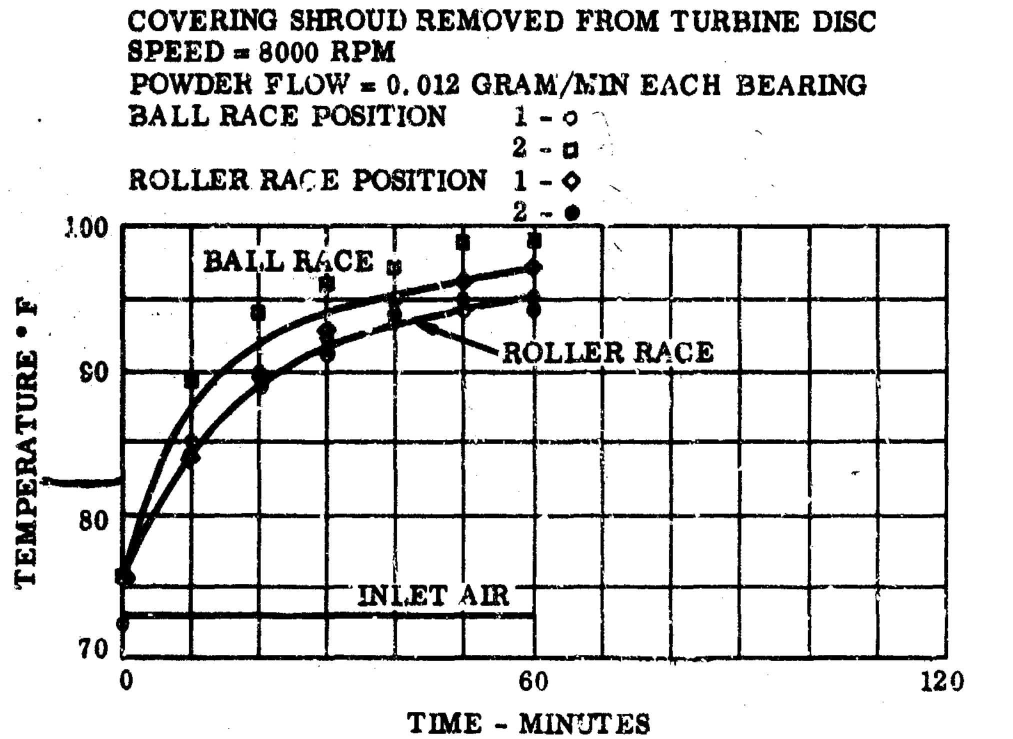

Figure 69. Engine Test E-63, Bearing Outer Race Temperature Versus Time

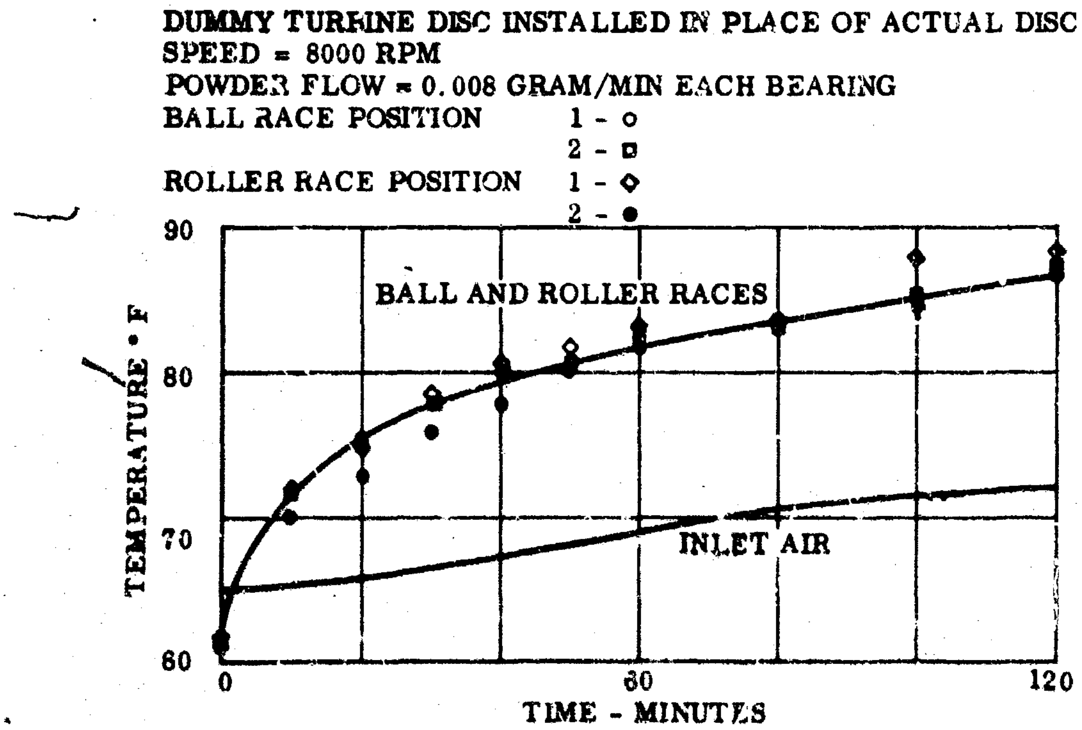

Figure 70. Engine Test E-64, Bearing Outer Race Temperature Versus Time

load, front bearing temperature stabilized at about 145° F, and with 50 pounds thrust, at about 300°F, which was over 100° F greater than previously obtained at similar conditions. Curves of bearing temperature versus time for this run are shown in figure 71.

There was found to be a possibility of thermocouple instrumentation malfunction during test E-65, and so after checkout of the instrumentation tests E-66 and E-67 were performed as checkout runs at 8000 rpm, with 0 to 50 pounds thrust load respectively. As shown in figures 72 and 73, ball bearing temperature stabilized at low levels for both of these runs, showing the bearing to evidently be in good running condition.

Test E-68 was run at 16,000 rpm with 50 pounds thrust load as a check with the previous test at these conditions (E-65). During this run ball bearing stabilization was reached at about 305° F, which was the same as during run E-65. After one hour of running the powder flow was increased from 0.010 to 0.033 grams per minute, causing a reduction of ball bearing temperature to about 270° F as shown in figure 74. At this higher flow condition the ball bearing temperature was slightly erratic, indicating that the higher flow rate was possibly causing some powder clogging in the bearing. Upon teardown after this run, there was seen to be a moderate amount of wear on the retainer guiding surfaces and wear tracks were ground into the outer ring guiding lands of the ball bearing.

Testing was continued using ball bearing S/N B-5, which has been used for previous testing at speeds to 12,000 rpm with no thrust load. Except for some burrs on the retainer, which were removed before installation, the bearing was in good condition, and performed well during the checkout test at 8000 rpm (test E-69, figure 75).

Test E-70 was run at 8000 rpm with 50 pounds thrust on the ball bearing. Ball bearing temperature during this run fluctuated between 145° F and 155° F, with some stabilization reached in the area of 155° F, as shown in figure 76. This run was stopped after 160 minutes because of excessive noise emission from the front bearing.

To determine if the bearing noise was indeed induced by thrust loading, test E-71 was started at 8000 rpm with no thrust load. After one hour, by which time the ball bearing operating temperature had stabilized in the area of 110° F, a thrust load of 50 pounds was applied, at which time ball bearing temperature increased to about 160° F, and the bearing became noisy. Curves of race temperature versus time for this test are shown in figure 77.

Upon engine teardown, bearing B-5 was seen to have retainer and ring guiding land wear similar to, although not as severe as, the previously run bearing (B-5c).

An inspection of the J-69 engine main bearing mounts at this time showed that the mounts were out of concentricity by 0.010 inch to 0.015 inch. This probable cause of the last two bearing failures was tracked back to the disintegration of the flywheel shroud following test E-62. During consultation with the manufacturer of the two failed bearings, this out of concentricity condition was confirmed as being the probable cause of the bearing failures.

Following engine bearing housing realignment to within 0.0005 inch and rebalancing of the rotor shaft, the engine was reassembled with new ball and roller

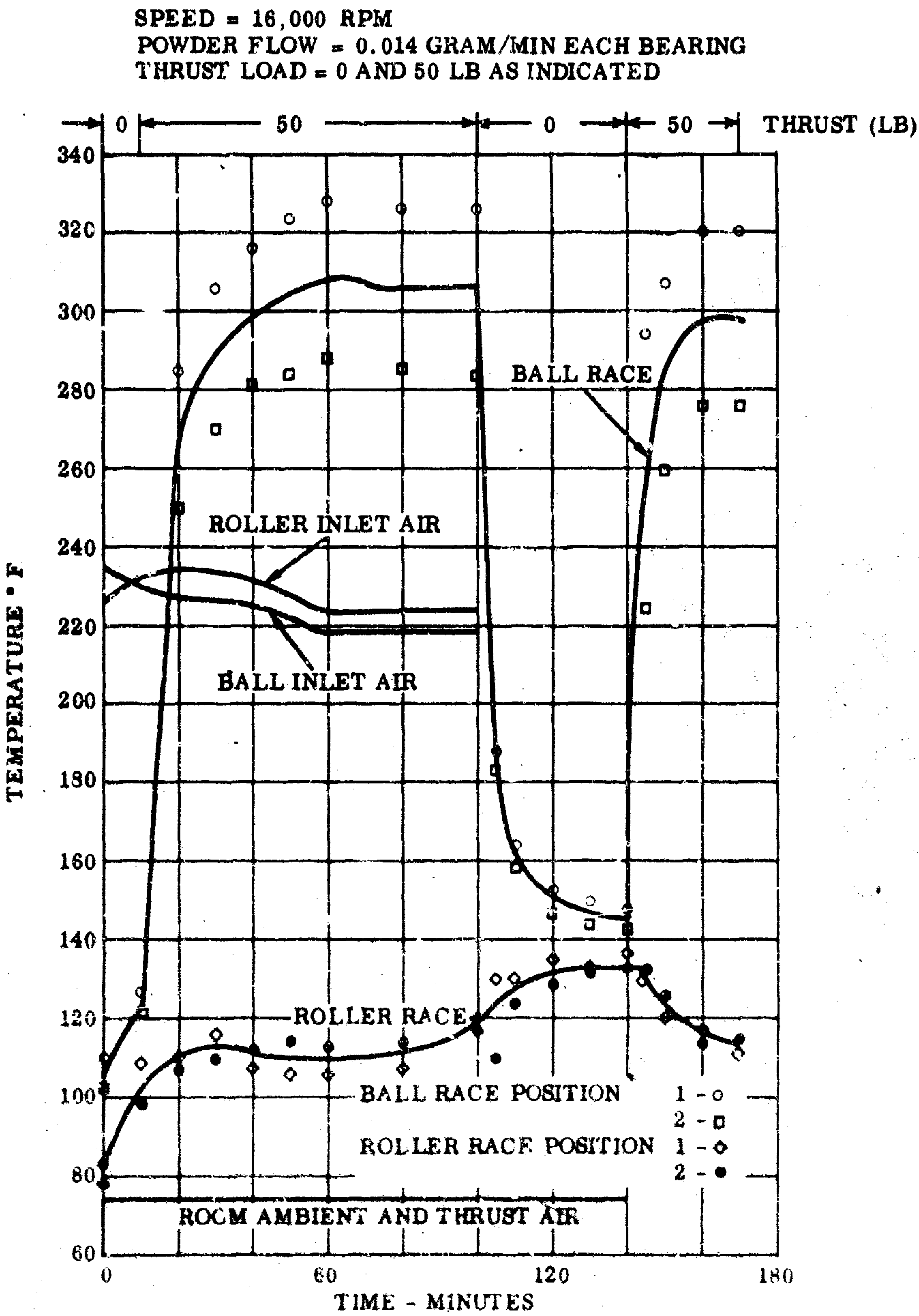

Figure 71. Engine Test E-65, Bearing Outer Race Temperature Versus Time

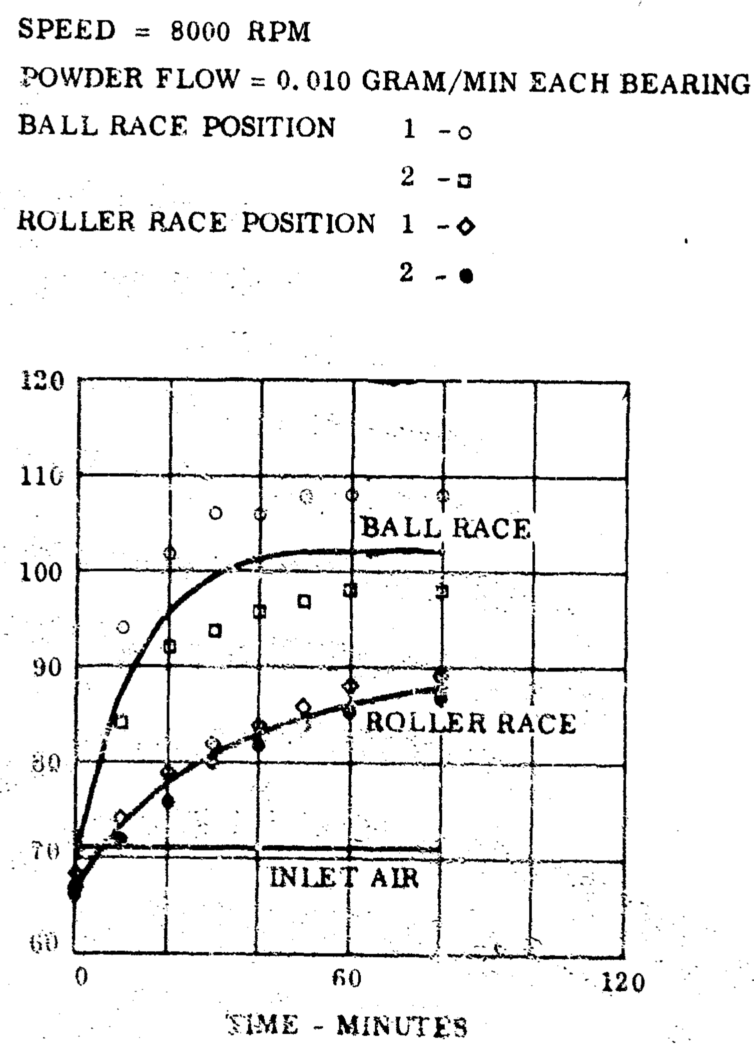

Figure 72. Engine Test E-66, Bearing Outer Race Temperature Versus Time

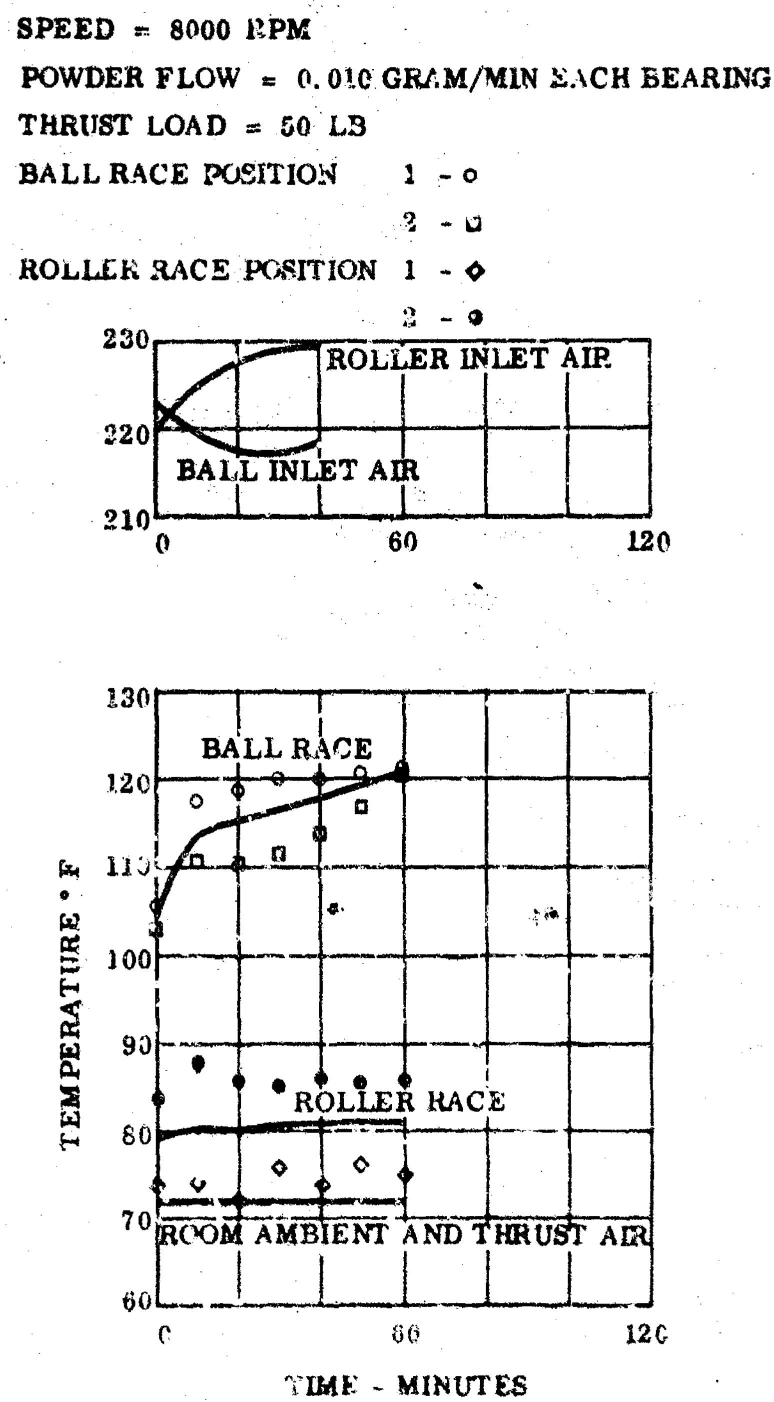

Figure 73. Engine [illegible] E-67, Bearing Outer Race Temperature Versus Time

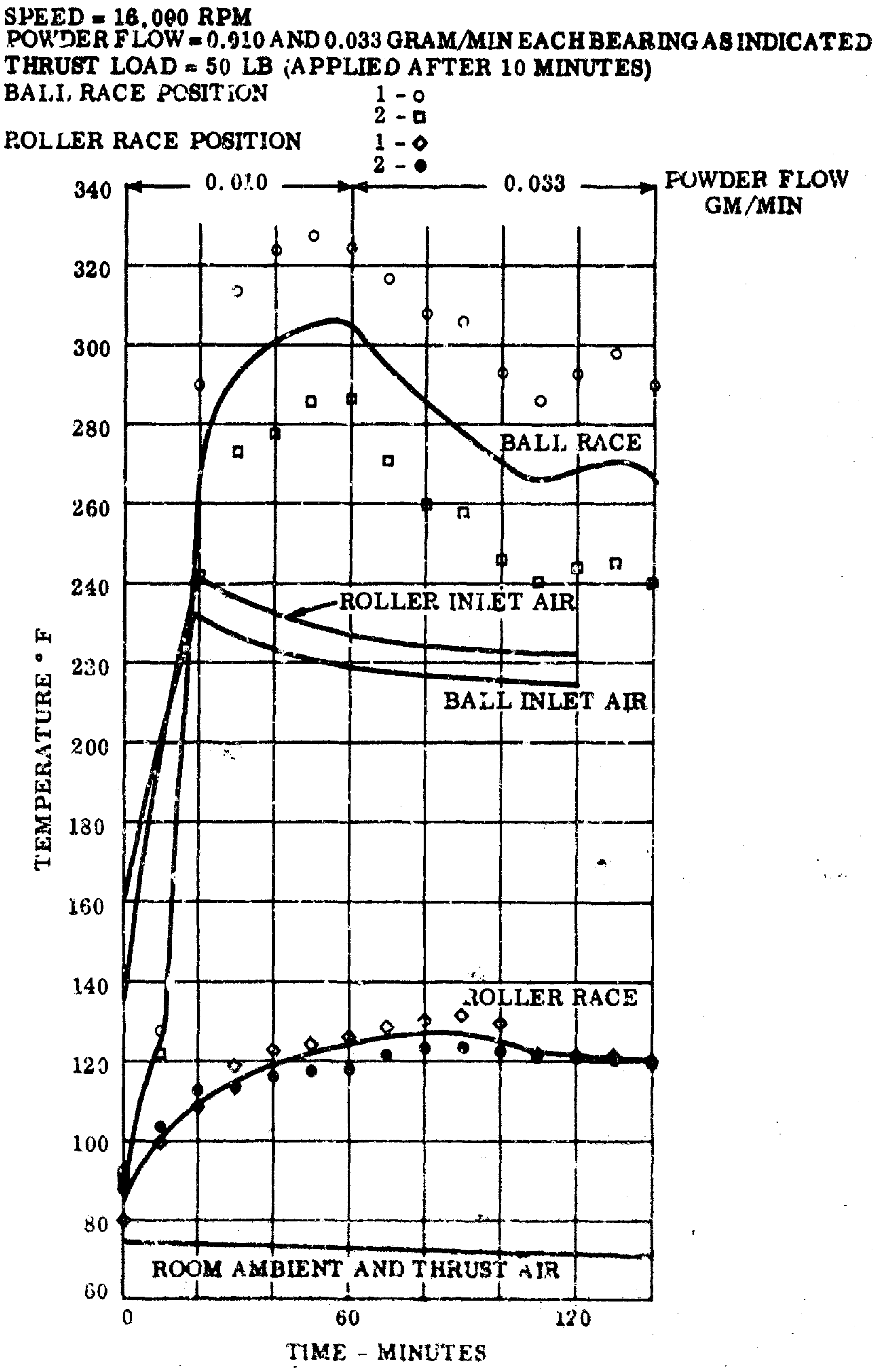

Figure 74. Engine Test E-68, Bearing Outer Race Temperature Versus Time

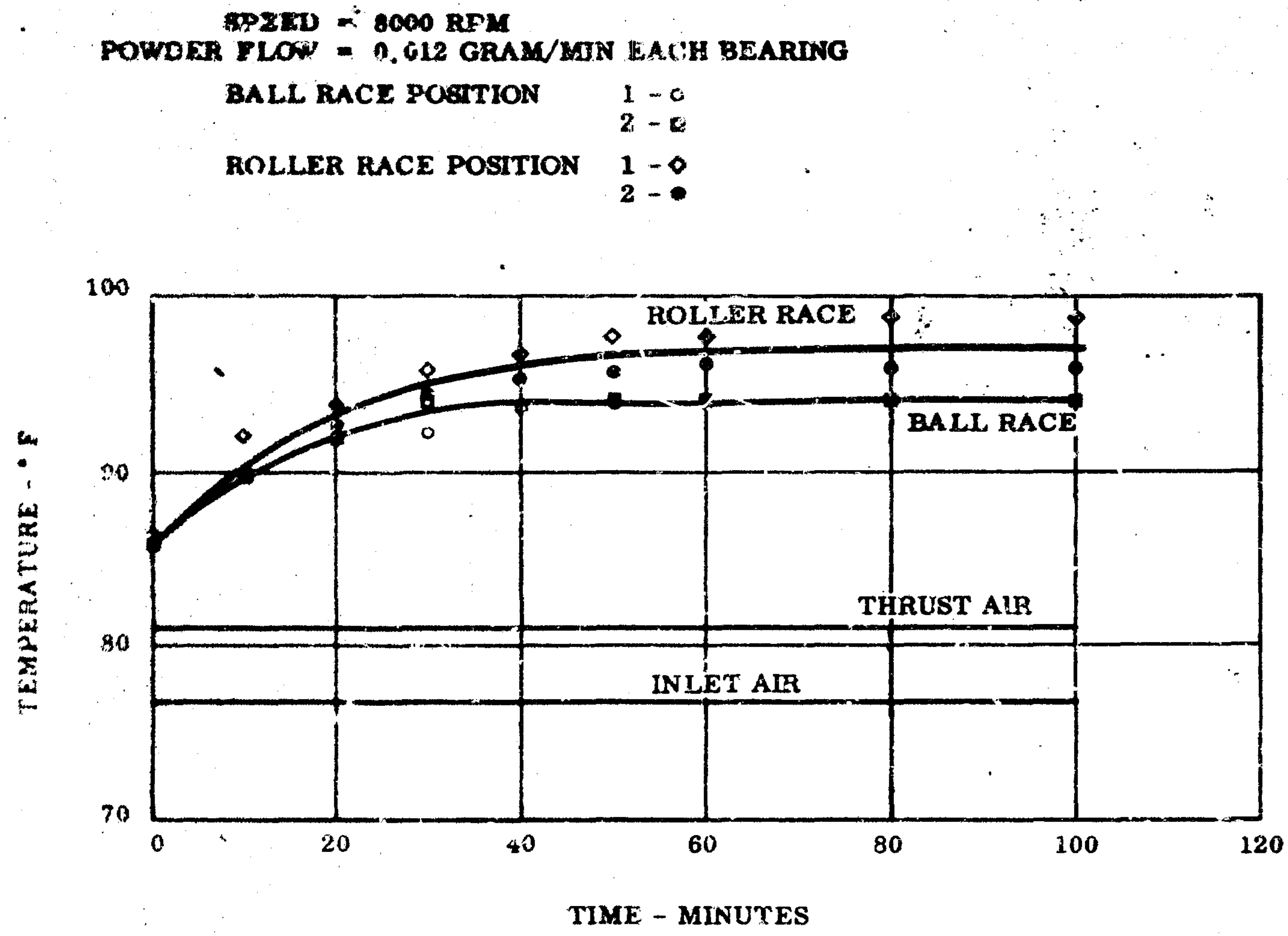

Figure 75. Engine Test E-69, Bearing Outer Race Temperature Versus Time

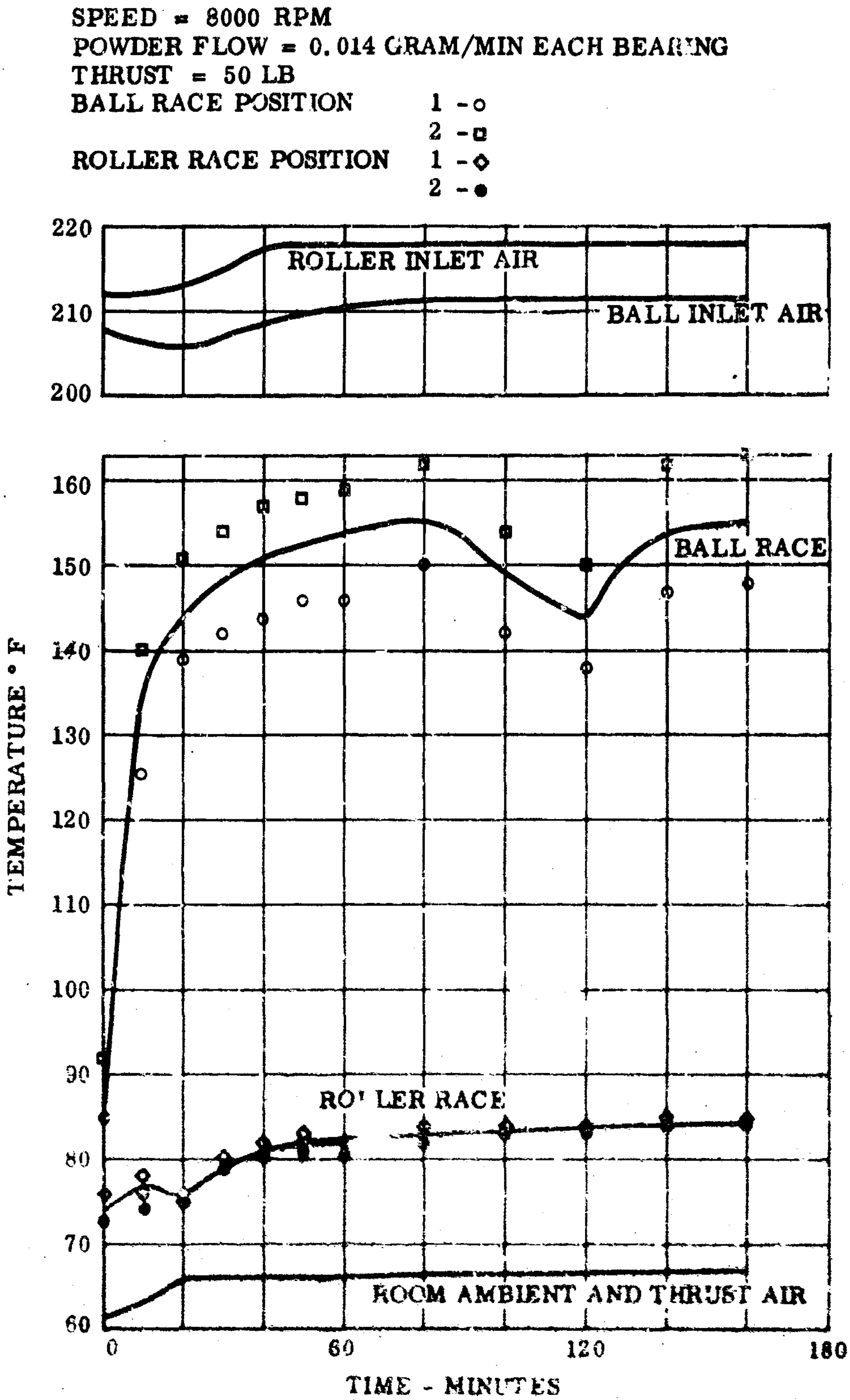

Figure 76. Engine Test E-70, Bearing Outer Race Temperature Versus Time

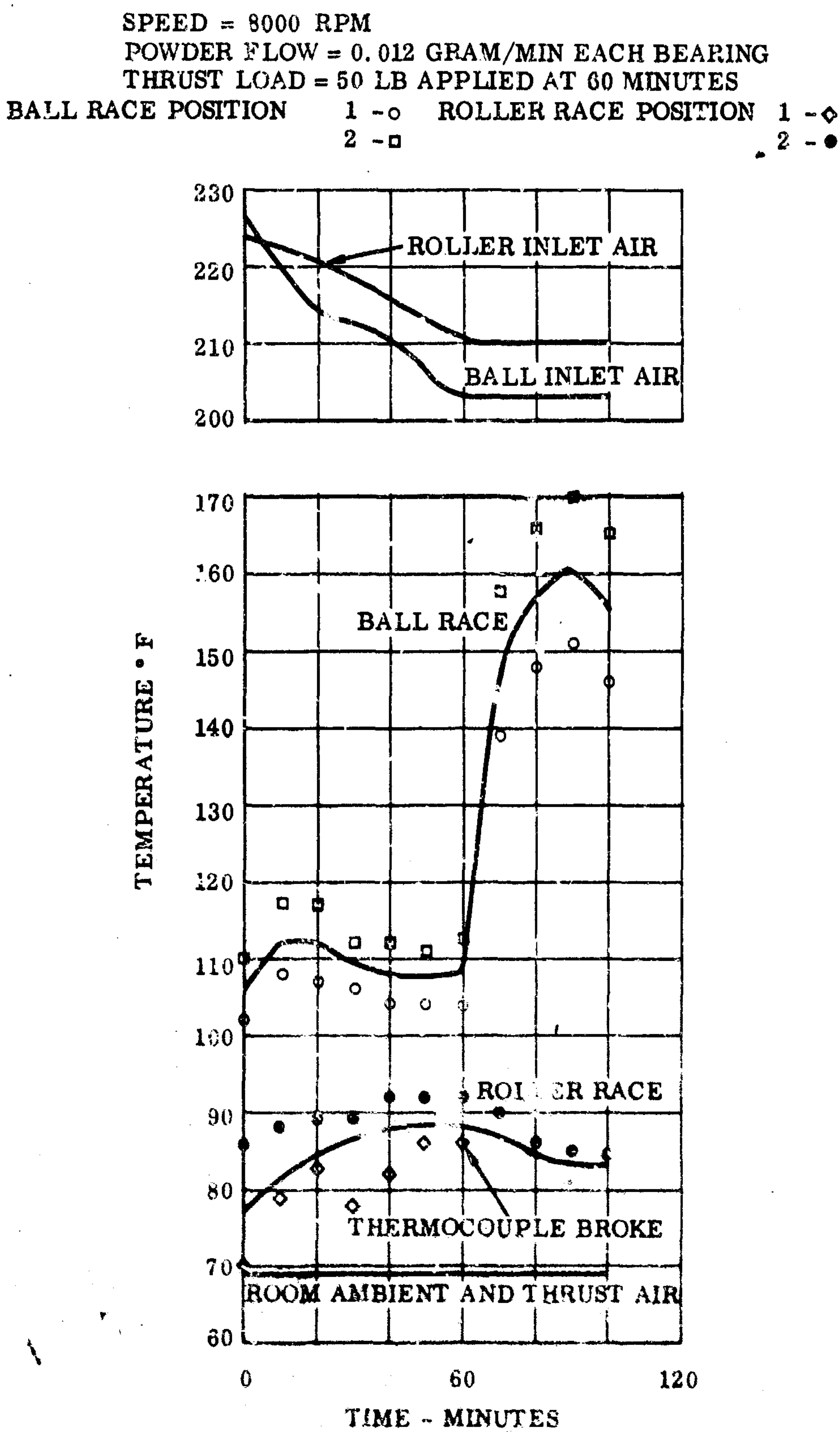

Figure 7. Engine Test E-71, Bearing Outer Race Temperature Versus Time

bearings (B-3a, R-1a) to be used for checking concentricity and balance. Test E-72 was started at 8000 rpm with no load and after 100 minutes a thrust load of 50 pounds was applied. Ball bearings stabilization temperature, as shown in figure 78 increased by about 10° F when the load was applied.

Test E-73 was run at 12,000 rpm, again with thrust loading increased from 0 to 50 pounds after one hour running. The bearings performed well during this run, as evidenced by the temperature stabilization as shown in figure 79.

Tests E-74 and E-75 were performed at 16,000 and 20,000 rpm, each run being started with zero thrust load which was raised to 50 pounds after one hour's running. As shown in figures 80 and 81, the bearings performed well at these speed and load levels. After one hour's running at 20,000 rpm with 50 pounds of thrust on the ball bearing, thrust load was increased to 100 pounds. Within five minutes, ball bearing temperature had increased from a stabilized value of about 190° F, to an unstable value of over 3000° F. After almost 30 minutes of unstable operation at this load level (shown in figure 81) the test was stopped because of this poor performance.

Following the testing at 20,000 rpm, a short run was made under 50 pounds thrust at 8000 and 12,000 rpm as a comparison with performance obtained during tests E-72 and E-73. Ball bearing temperatures were about 15° F to 20° F higher than during the earlier tests, and the ball bearing sounded noisy, so this test was stopped and the engine was torn down. On disassembly, a shaft locking nut at the front bearing assembly was found to be loose on the shaft, and it was felt that this looseness was the probable cause of erratic bearing operation at the high thrust load condition. The ball bearing itself was found to be in good condition, with a good lubricant film on all surfaces and only some light retainer wear.

While erratic running conditions were encountered at high speed - high load operations, a complete series of runs at varying speeds and loads had been made with ball bearing B-3a. The results of this testing, plotted as temperature-speed curves with thrust load as parameter, are presented in figure 30.

For continued testing, the R-1a roller bearing was kept in the engine and a new ball bearing S/N B-5d was installed. This differed from previous B-5 series bearings in that no powder exhaust grooves were cut in the retainer. (B-5 series is the design tentatively selected for use in the final engine design). Test E-76 was a checkout run of the new bearing and was started with no thrust load, a load of 50 pounds being applied after 90 minutes. Temperature stabilization was readily achieved under both loading conditions, as shown in figure 82.

The next test run was started at 12,000 rpm but was shut down when excessive vibration was found in the engine. Subsequent testing and inspection revealed that this vibration was caused by an out of balance condition in the rotor shaft, and the shaft was rebalanced, then the engine was reassembled with new bearings (B-4a and R-4).

Test E-77 was performed with these new bearings as a check of the engine assembly balance at 8000 rpm with 50 pounds thrust load. Bearing stabilization was readily achieved as shown in figure 83, indicating that the shaft was in balance. A previous test, E-53 with this bearing failed in a similar manner; the failure was in the retainer design and not the Monel S material.

The next test was started at 12,000 rpm but was shut down when ball bearing temperature increased rapidly three minutes after applying 50 pounds thrust load.

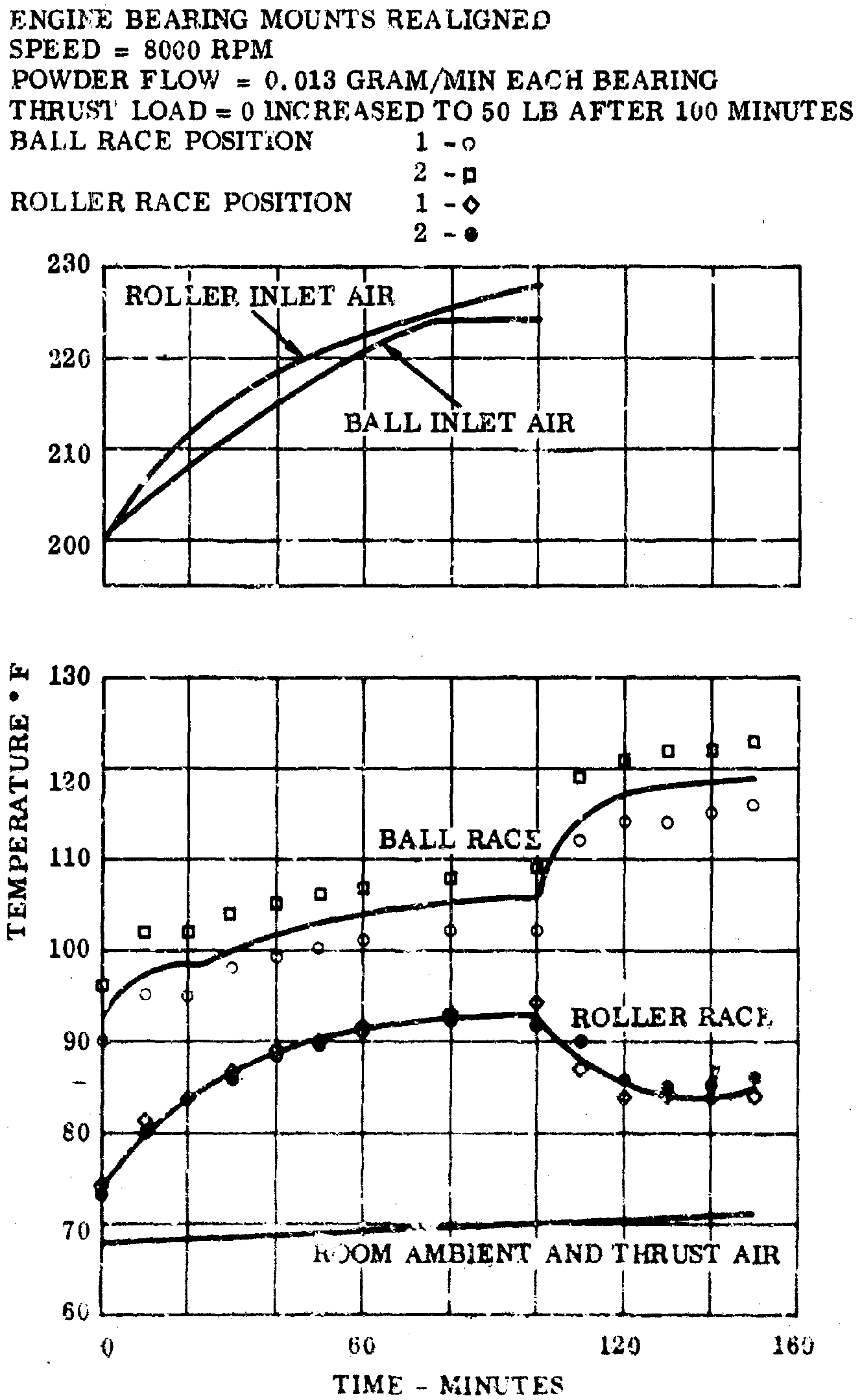

Figure 78. Engine Test E-72, Bearing Outer Race Temperature Versus Time

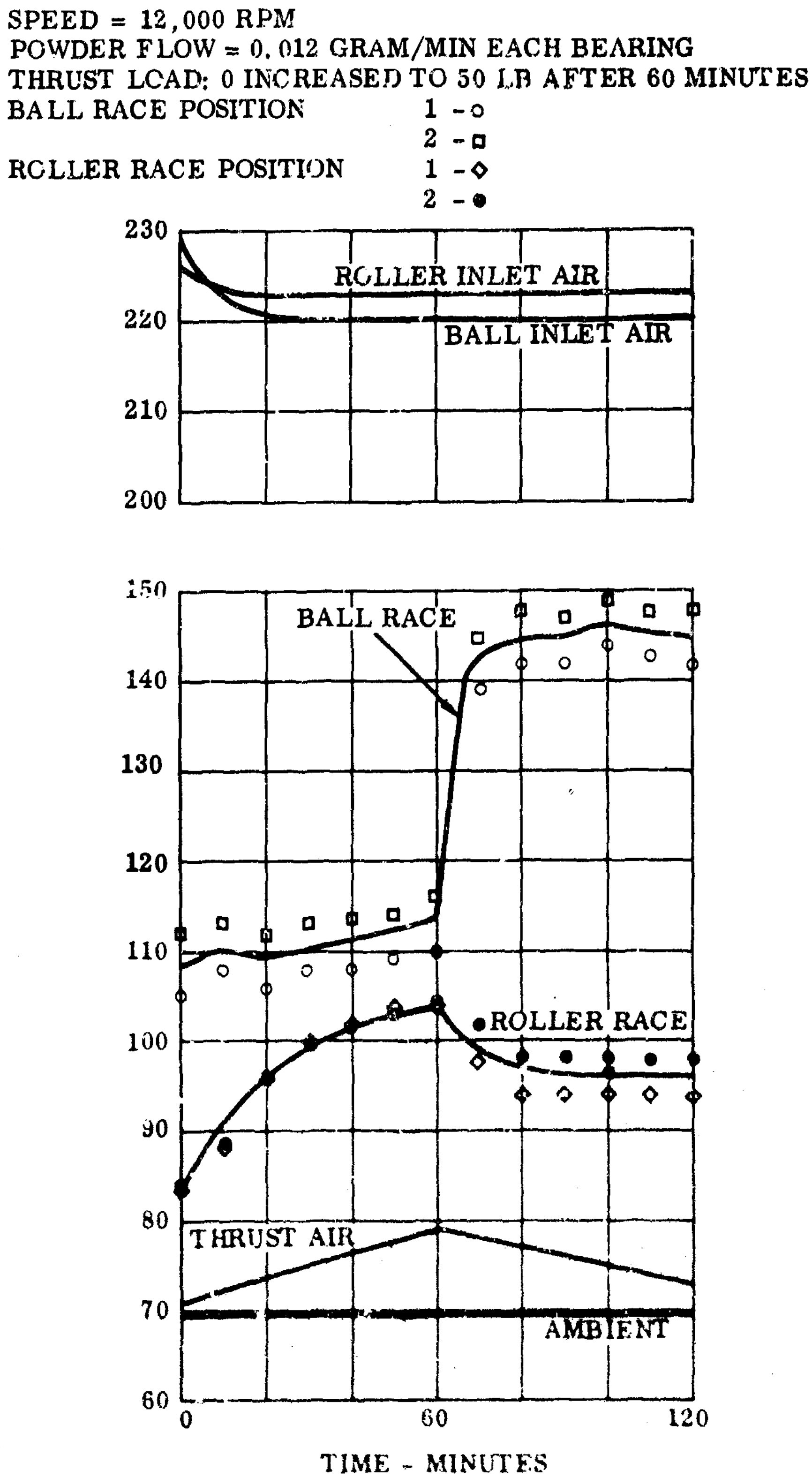

Figure 79. Engine Test E-73 Bearing Outer Race Temperature Versus Time

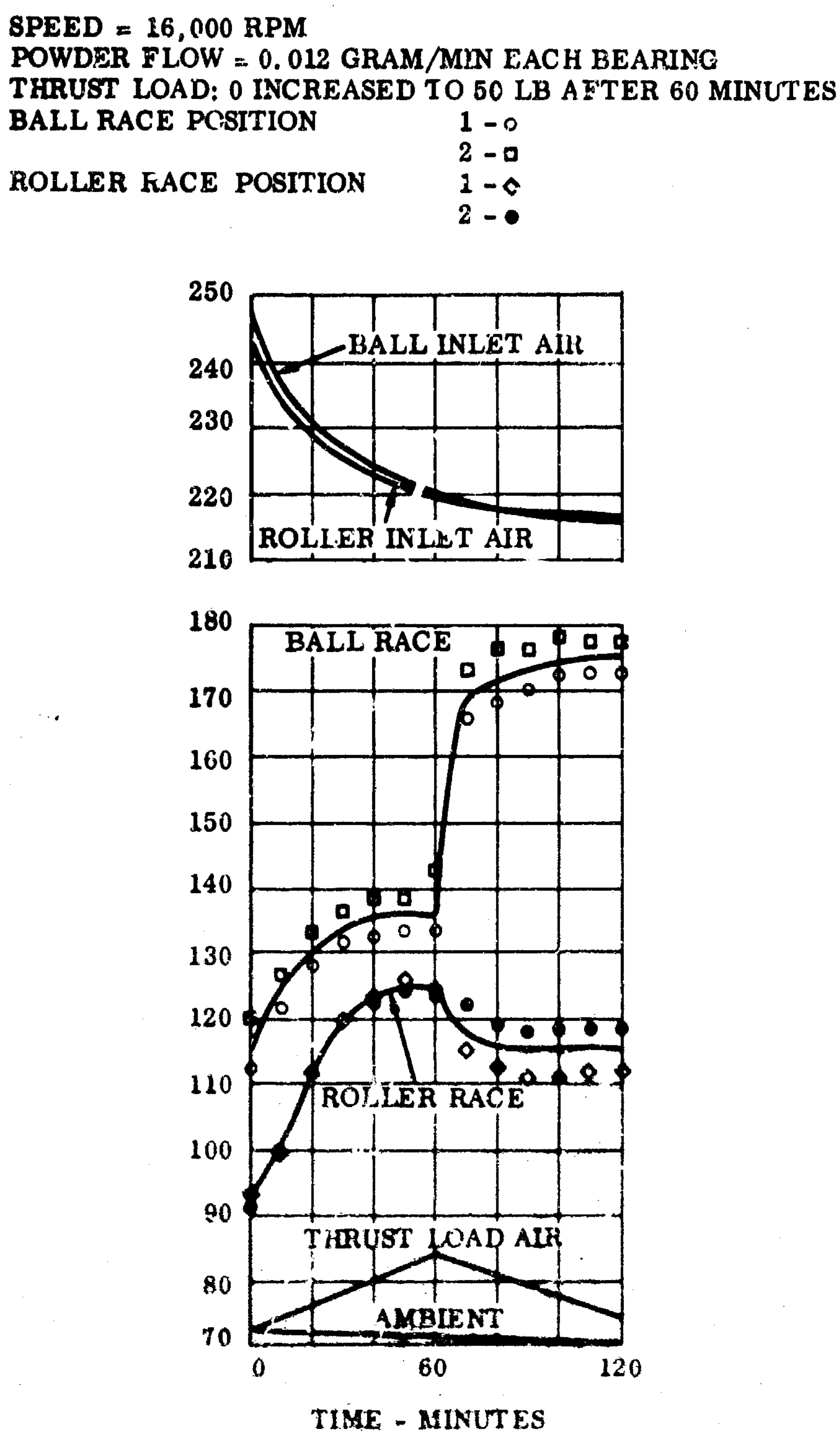

Figure 80. Engine Test E-74, Bearing Outer Race Temperature Versus Time

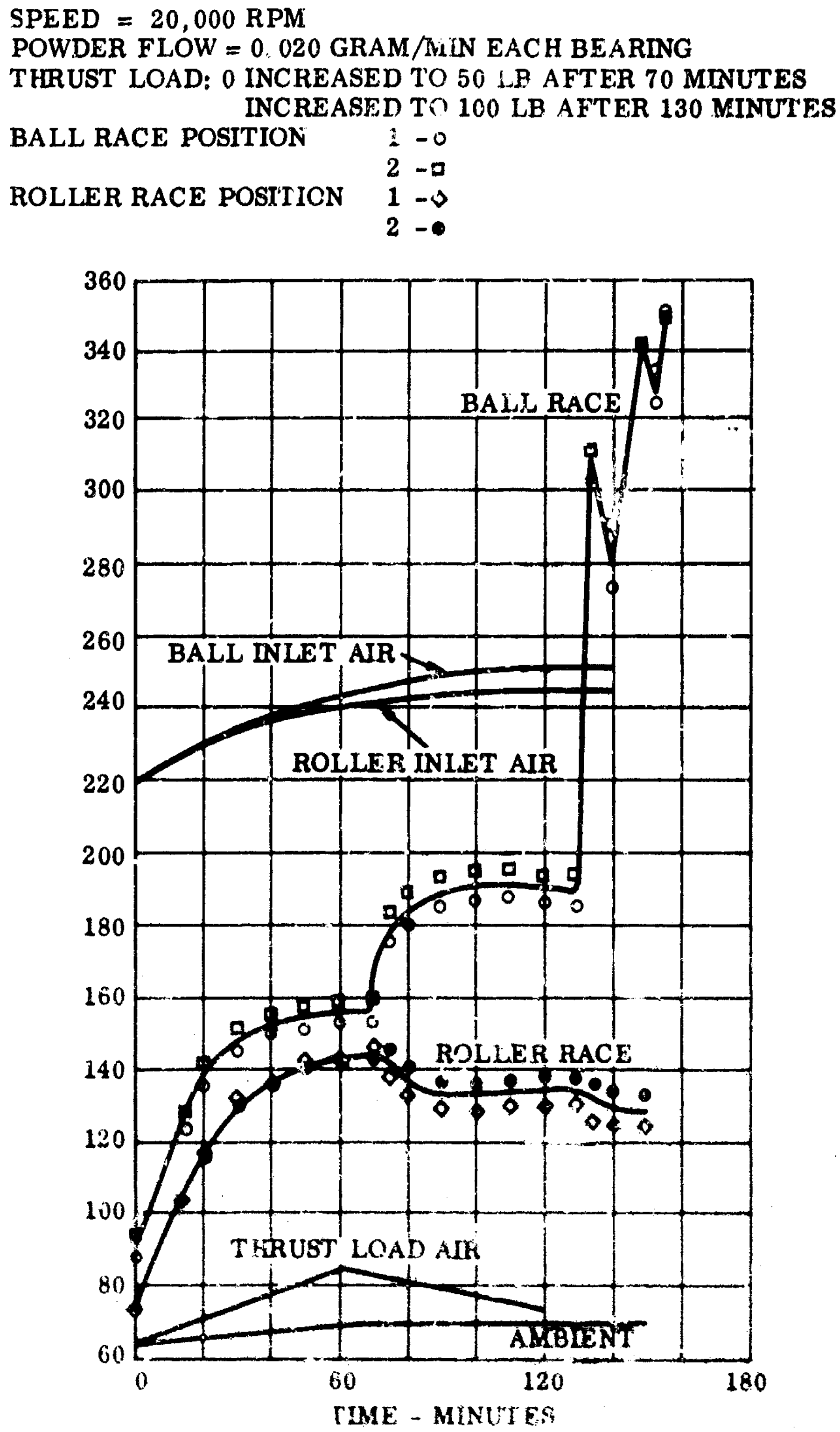

Figure 81. Engine Test E-75, Bearing Outer Race Temperature Versus Time

SPEED = 8000 RPM
POWDER FLOW = 0.015 GRAM/MIN EACH BEARING
THRUST LOAD: 0 INCREASED TO 50 LB AFTER 10 MINUTES
BALL RACE POSITION 1 - o
2 - □
ROLLER RACE POSITION 1 - ◇
2 - ●

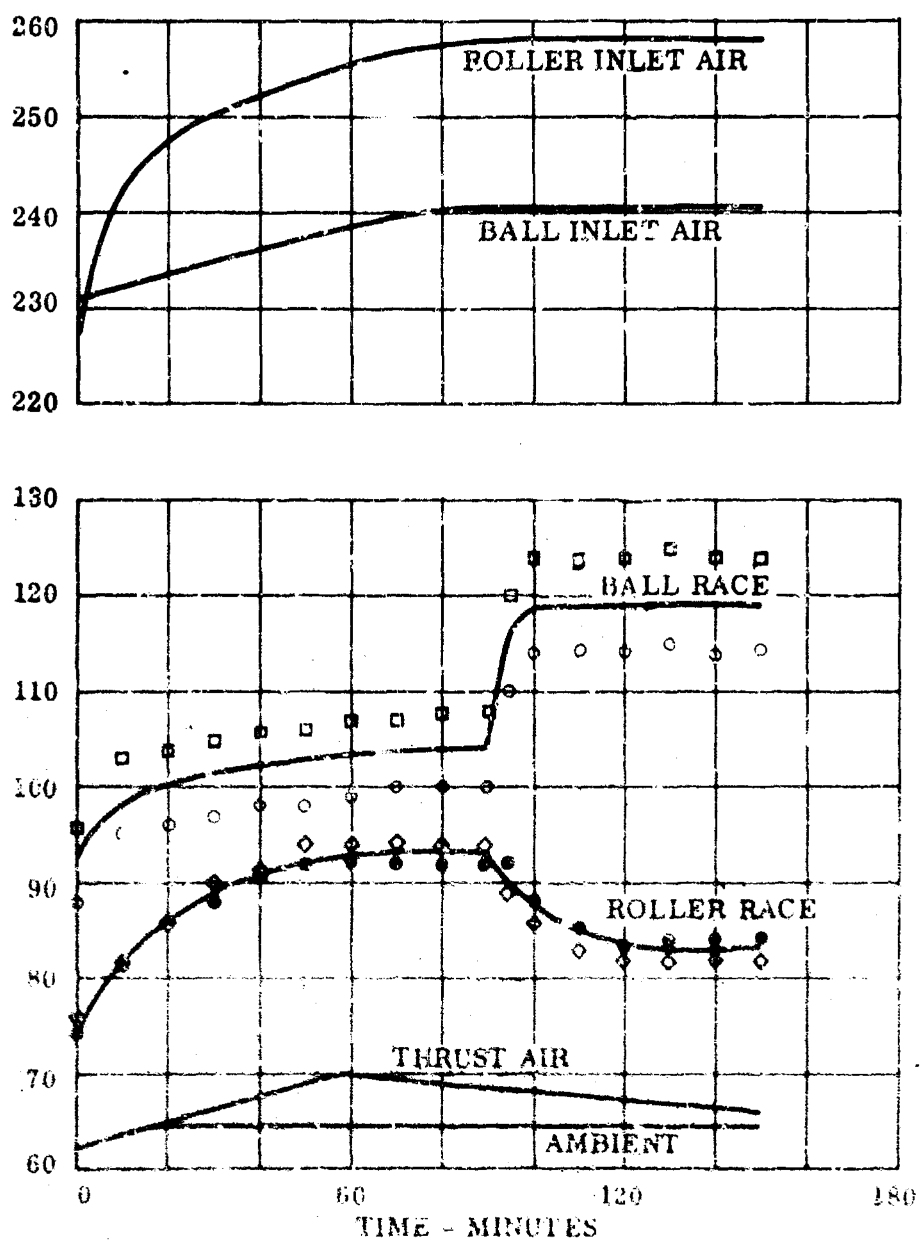

Figure 82. Engine Test E-76, Bearing Outer Race Temperature Versus Time

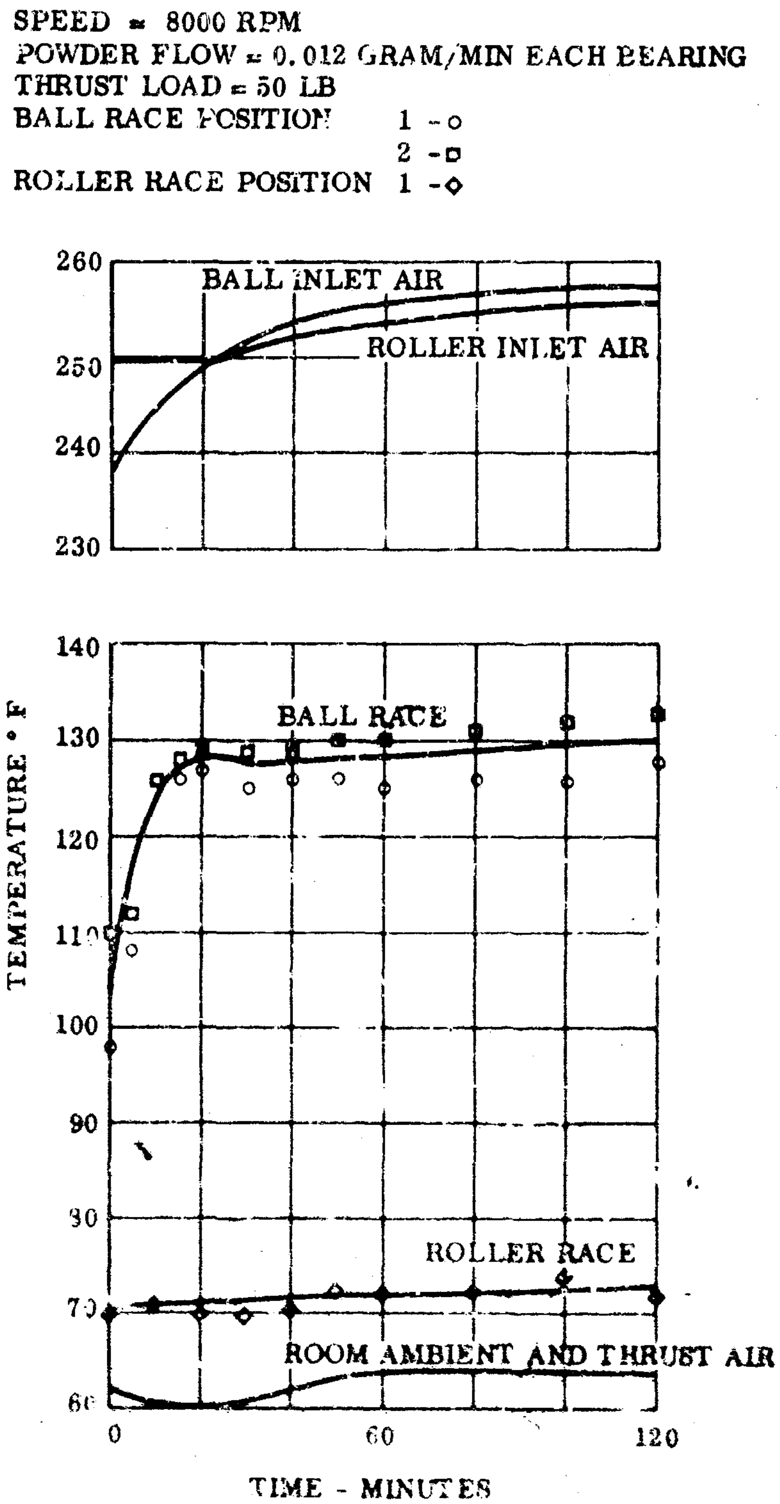

Figure 83. Engine Test E-77, Bearing Outer Race Temperature Versus Time

Inspection revealed lack of powder lubricant filming on the Monel retainer to be the cause of failure.

For test E-78, roller bearing R-4 was kept in the engine, and ball bearing B-5d was reinstalled. This test was performed at 8000 rpm as a check with test E-76. Bearing stabilization was achieved at approximately the same temperatures as the previous test, as shown in figure 84.

Test E-79 was performed at 12,000 rpm with 50 pounds thrust load, and while bearing temperature stabilization was achieved as shown in figure 85, the front bearing sounded somewhat noisier than usual. The next test run was started at 16,000 rpm with 50 pounds thrust load but was shut down after a few minutes because of high torque and noise level at the front bearing. Disassembly showed the bearing to be in good condition with a good lubricant film on balls and races, but an excessive amount of retainer wear. The retainer was deburred and powder exhaust grooves were cut in it prior to reassembly of the bearing in the engine.

Test E-80 was a low speed (8000 rpm) checkout run of the engine with the now modified B-5d ball bearing. The bearings ran well, as shown in figure 86, and the ball bearing stabilized at a slightly lower temperature than during test E-76.

Test E-81 was run at 16,000 rpm with no thrust load, but difficulties with the powder flow system during this run invalidated the data obtained.

Test E-82 was performed at 12,000 rpm with no thrust load but with powder flow varied by varying lubricator feedwheel speed. There was seen to be a negligible change in temperature of both the ball and roller bearings as this feedwheel speed was increased from its normal 2 rpm to a maximum of 10 rpm. Curves of race temperature versus time are shown in figures 87 and 88, (this test was run in two parts).

Test E-83 was performed to determine the effect of changing lubricant flow with a bearing speed of 16,000 rpm. Unlike the results obtained at 12,000 rpm, there was a drop in ball bearing temperature of about 145° F when the lubricator feedwheel speed was increased from 2 to 4 rpm. There was no significant change in ball bearing temperature as feedwheel speed was increased to 6 rpm, indicating that there is evidently a "critical threshold flow" at which point powder lubrication shows a marked increase in effectiveness. This might be due to windage effects in the bearing, or possibly to the formation of a more thorough temporary powder coating on the bearing surfaces as powder flow is increased beyond the required minimum level. It is interesting to note that as lubricator speed was increased from 2 to 4 rpm, there was a slight increase in roller bearing temperature, in this case possibly caused by the powder flow being sufficient to cause a slight amount of clogging (caking) in the roller bearing. Curves of race temperature versus time for run E-83, which was performed in two parts, are shown in figures 89 and 90.

Test E-84 was performed at 20,000 rpm with a thurst load of 50 pounds applied after 110 minutes. This test was performed with powder flow at the higher level shown to be desirable during the previous run (estimated at about 0.025 grams per minute). There was a sharp increase in ball bearing stabilization temperature (about 35° F) when the thrust load was applied, but after 30 minutes running with load, ball bearing temperature dropped gradually and restabilized at about 182° F as shown in figure 91. The drop in temperature was probably caused by caked powder slowly

SPEED = 8000 RPM
POWDER FLOW = 0.012 GRAM/MIN EACH BEARING
THRUST LOAD = 50 LB
BALL RACE POSITION 1 - ○
2 - □
ROLLER RACE POSITION 1 - ◇

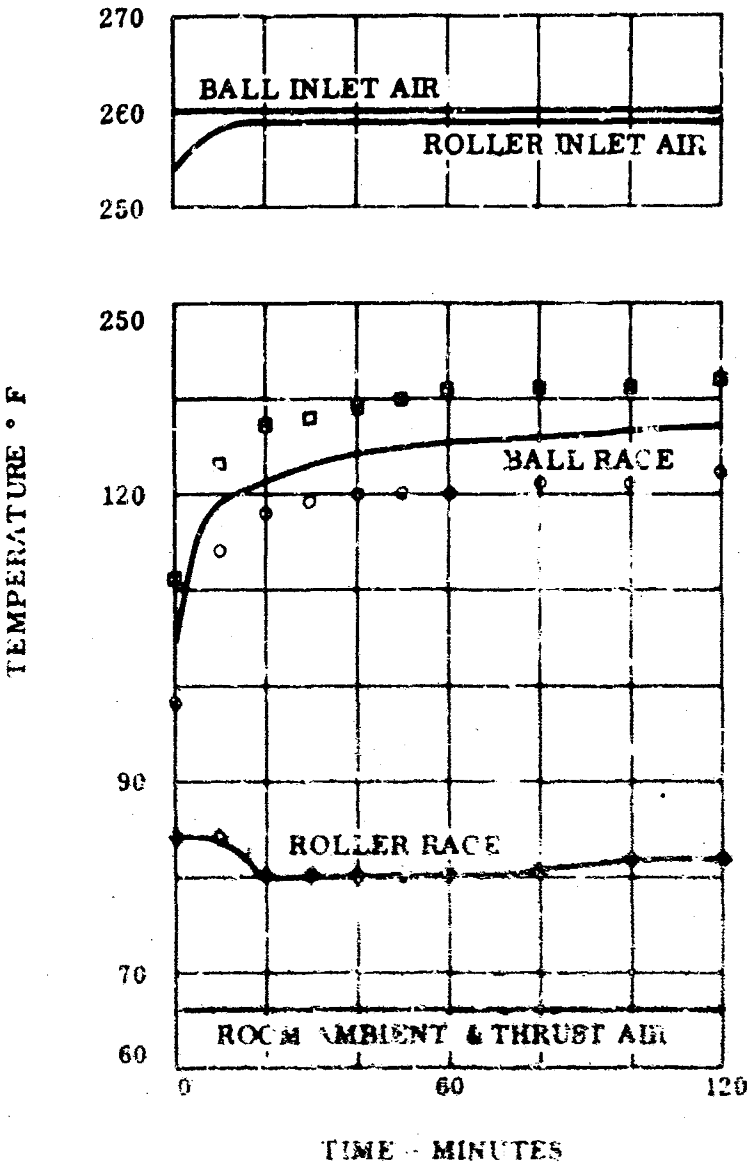

Figure 84. Engine Test E-78, Bearing Outer Race Temperature Versus Time

SPEED = 12,000 RPM
POWDER FLOW = 0.013 GRAM/MIN EACH BEARING
THRUST LOAD = 50 LB
BALL RACE POSITION 1 - ○
2 - □
ROLLER RACE POSITION 1 - ◇

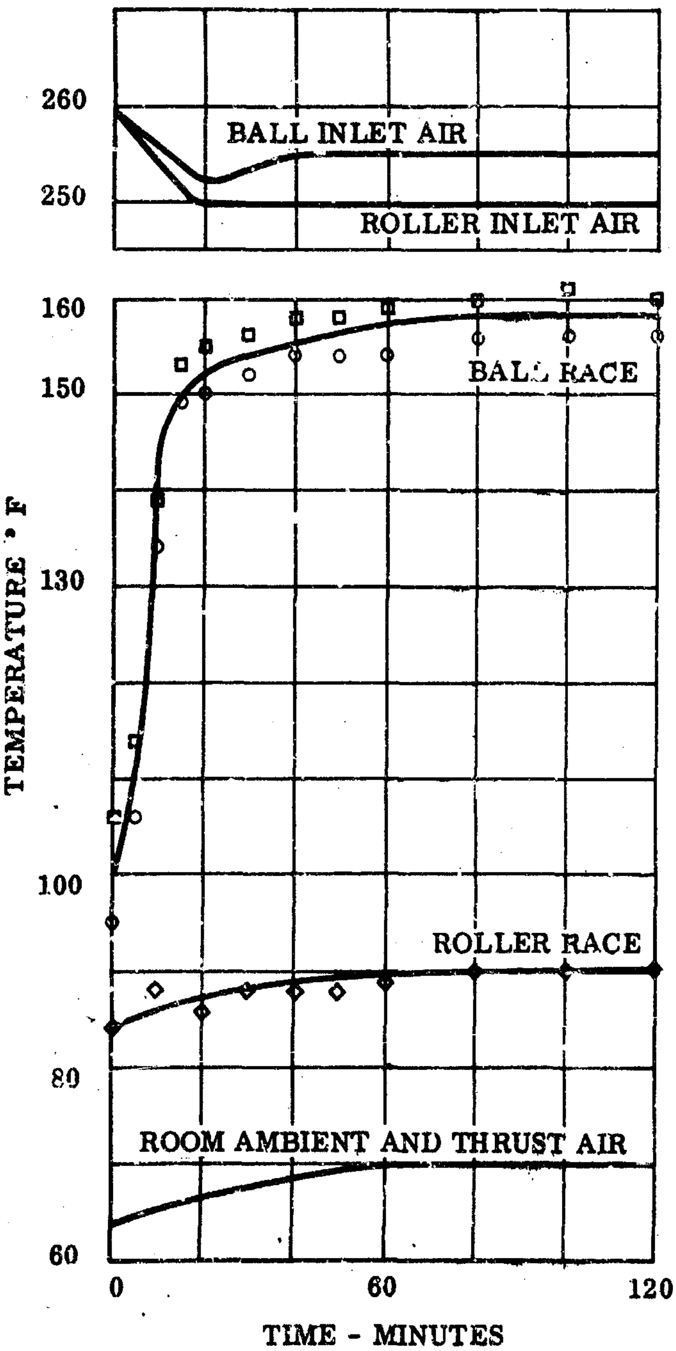

Figure 55. Engine Test E-79, Bearing Outer Race Temperature Versus Time

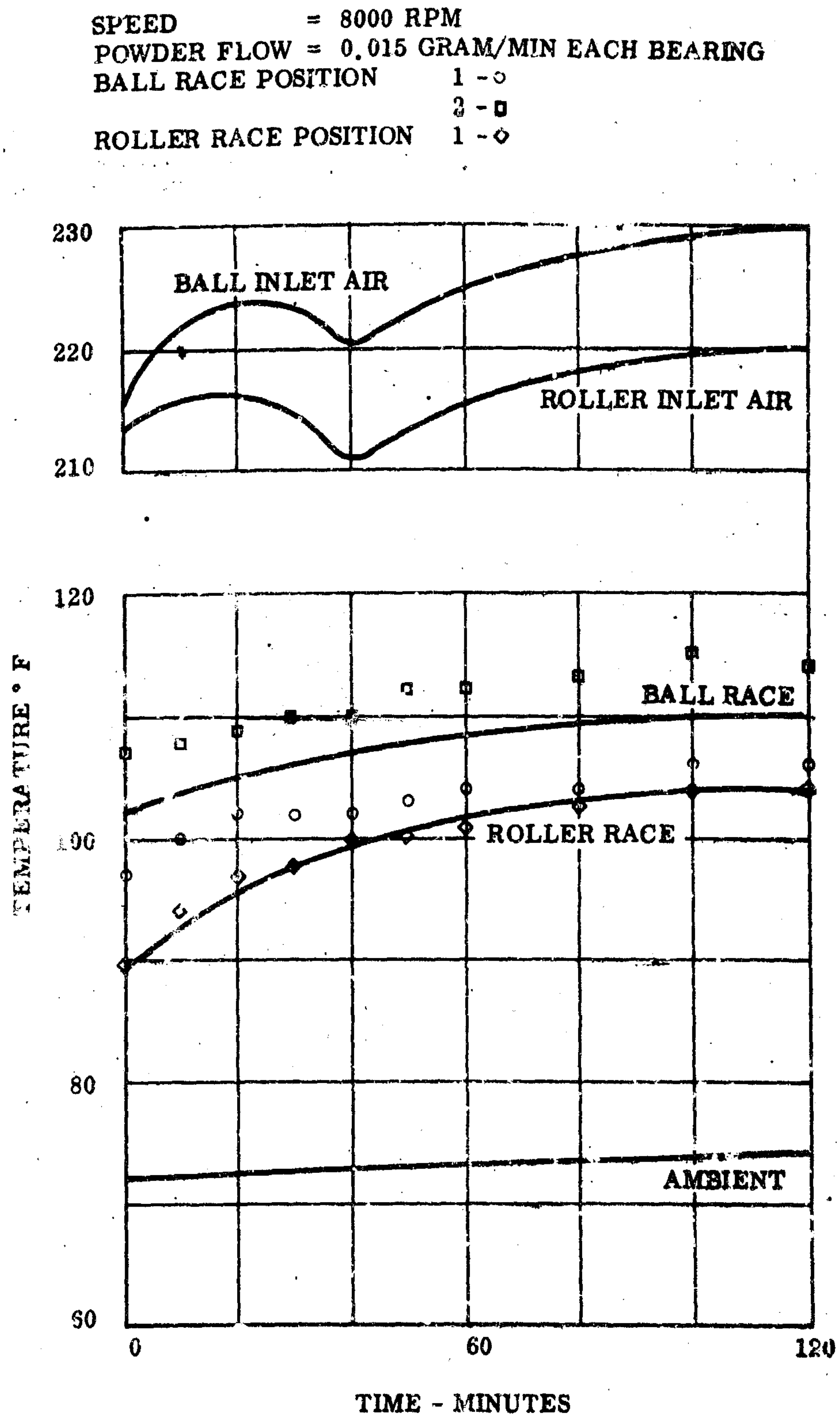

Figure 86. Engine Test E-80, Bearing Outer Race Temperature Versus Time

SPEED = 12,000 RPM
POWDER FLOW - VARIED, LUBRICATOR FEEDWHEEL SPEED AS INDICATED, NORMAL SPEED IS 2 RPM
BALL RACE POSITION 1 - ○
2 - □
ROLLER RACE POSITION 1 - ◇

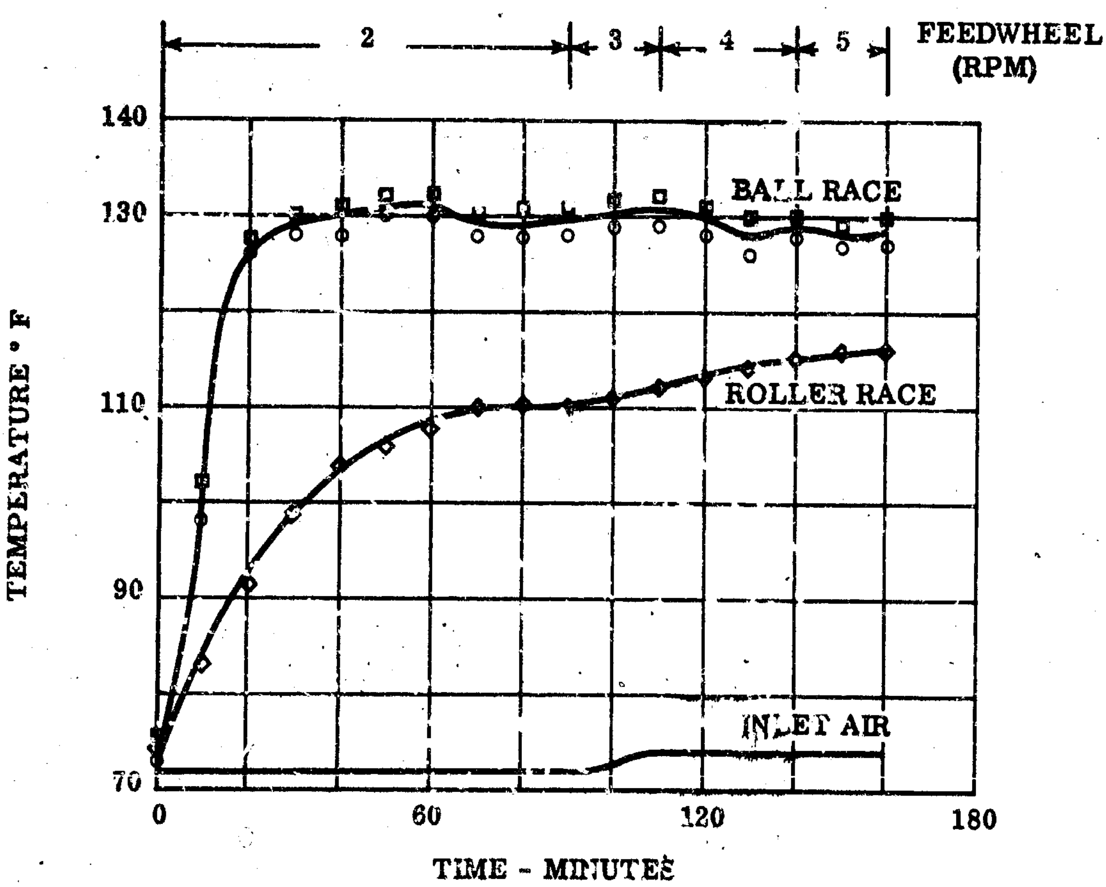

Figure 87. Engine Test E-82, Bearing Outer Race Temperature Versus Time

SPEED = 12,000 RPM
POWDER FLOW VARIED, LUBRICATOR FEEDWHEEL SPEED AS INDICATED, NORMAL IS 2 RPM
BALL RACE POSITION 1 - ○
2 - □
ROLLER RACE POSITION 1 - ◇

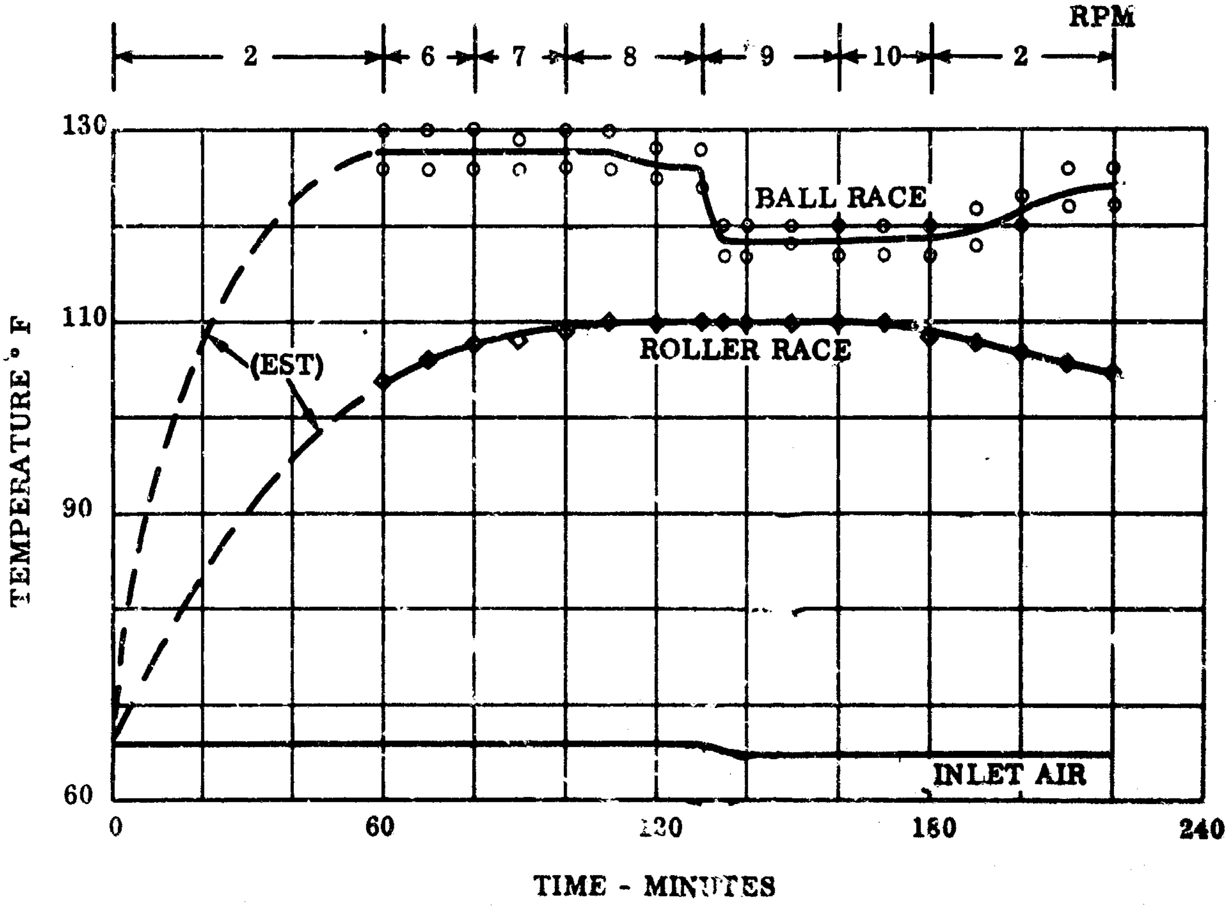

Figure 88. Engine Test E-82 (Part II), Bearing Outer Race Temperature Versus Time

SPEED = 16,000 RPM
POWDER FLOW - VARIES, FEEDWHEEL SPEED AS INDICATED, NORMAL IS 2 RPM
BALL RACE POSITION 1 - ○
2 - □
ROLLER RACE POSITION 1 - ◇

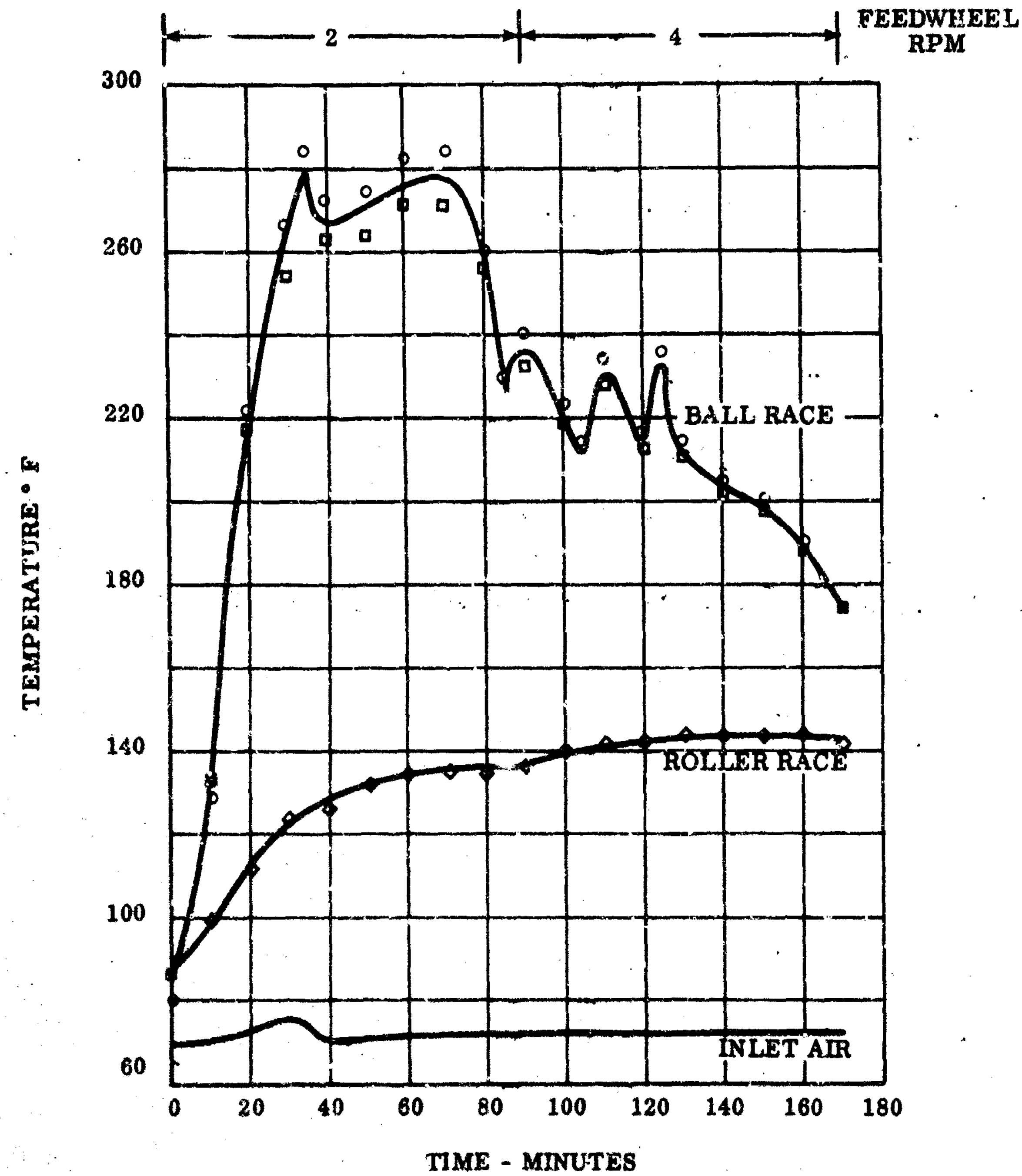

Figure 89. Engine Test E-83, Bearing Outer Race Temperature Versus Time

SPEED = 10,000 RPM
POWDER FLOW - VARIES; FEEDWHEEL SPEED AS INDICATED; NORMAL IS 2 RPM
BALL RACE POSITION 1 - ○
2 - □
ROLLER RACE POSITION 1 - ◇

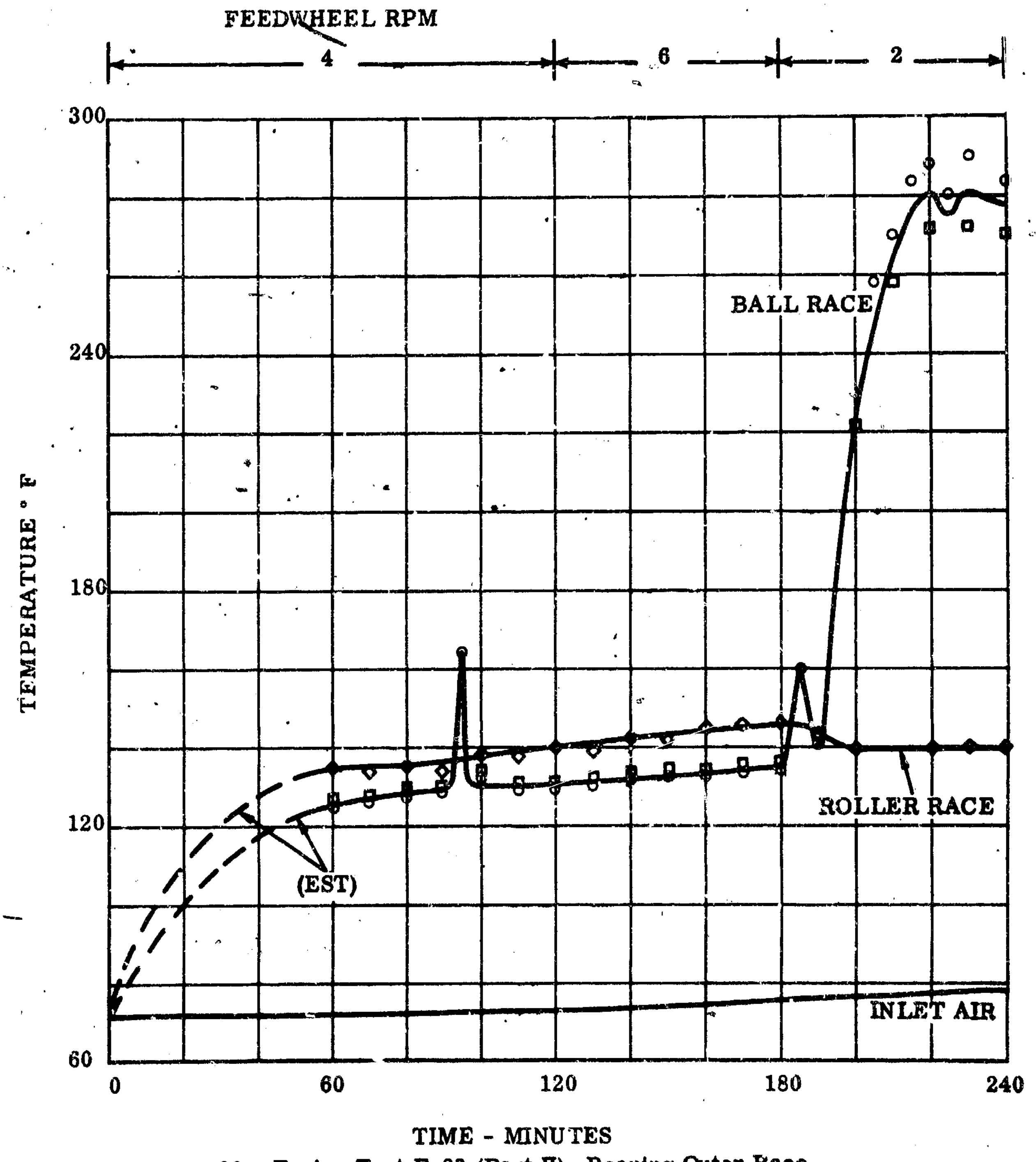

Figure 90. Engine Test E-83 (Part II), Bearing Outer Race Temperature Versus Time

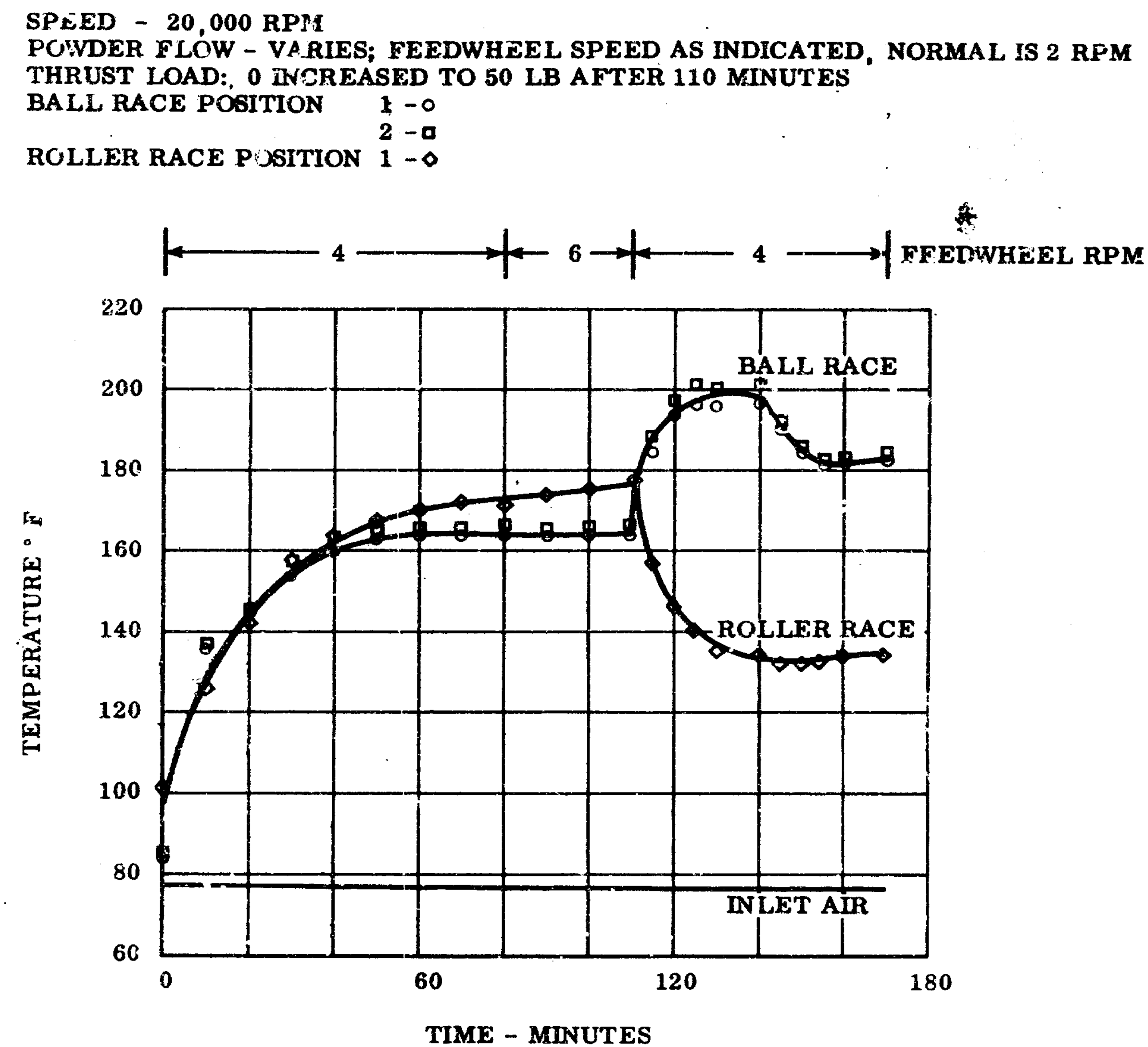

Figure 91. Engine Test E-84, Bearing Outer Race Temperature Versus Time

wearing out of the bearing subsequent to its being run with a lubricator feedwheel speed of 6 rpm.

Test E-85 was performed with conditions similar to the previous run, but with lubricant carrier air heated to simulate engine bleed air temperature. Ball bearing temperature stabilized at 200° F, but after about 1-1/2 hours, the temperature began to rise, until it reached approximate stabilization, fluctuating around 250° F, as shown in figure 92. This slight fluctuation of plus or minus 5° F was not considered detrimental to bearing operation, and it is probable that increase in operating temperature after 1-1/2 hours was caused by slight flaking of powder or silver burrs from the retainer, which nevertheless did not prevent the bearing from performing adequately. This test marked the second ball bearing, and the second bearing desing, which performed successfully at 20,000 rpm under simulated engine thrust load conditions, demonstrating the degree of reliability which has been achieved with powder lubrication at this time. Curves showing speed versus stabilization temperature (with thrust load as parameter) for the B-5 series bearings, including previous testing with bearings B-5 and B-5c, are shown in figure 31.

Test E-86 was performed successfully at 20,000 rpm with a thrust load of 75 pounds on the ball bearing. With a lubricator feedwheel speed of 4 rpm (estimate powder flow of 0.025 grams per minute each bearing), ball bearing temperature stabilized at 300° F. When feedwheel speed was increased to 8 rpm, bearing temperature began to drop slightly, showing indications of stabilizing at about 270° F. It was felt, however, that there was too much of a risk of powder caking in the bearing to warrant continued testing at this high flow rate. During this test roller bearing temperature did not stabilize but fluctuated between 100° F and 130° F, possibly due to some powder caking in the bearing. Curves of race temperature versus time are shown in figure 93.

Test E-87 was run at 20,000 rpm with about 90 pounds thrust load on the ball bearing with a powder flow of 0.025 grams per minute each bearing (4 prm feedwheel speed). The ball bearing showed a moderate degree of temperature stabilization at about 270° F, and the roller bearing stabilized at 90° F as shown in figure 94.

It should be noted that with the successful completion of test E-87, ball bearing B-5d had accomplished about 35 hours of running, including over 8-1/2 hours at 20,000 rpm with 50 to 100 pounds thrust load.

SPEED = 20,000 RPM
POWDER FLOW - VARIES, FEEDWHEEL SPEED AS INDICATED (NORMAL 2 RPM
THRUST LOAD - 50 POUNDS APPLIED AT 20 MINUTES
BALL RACE POSITION 1 - o
2 - □
ROLLER RACE THERMOCOUPLES INOPERATIVE

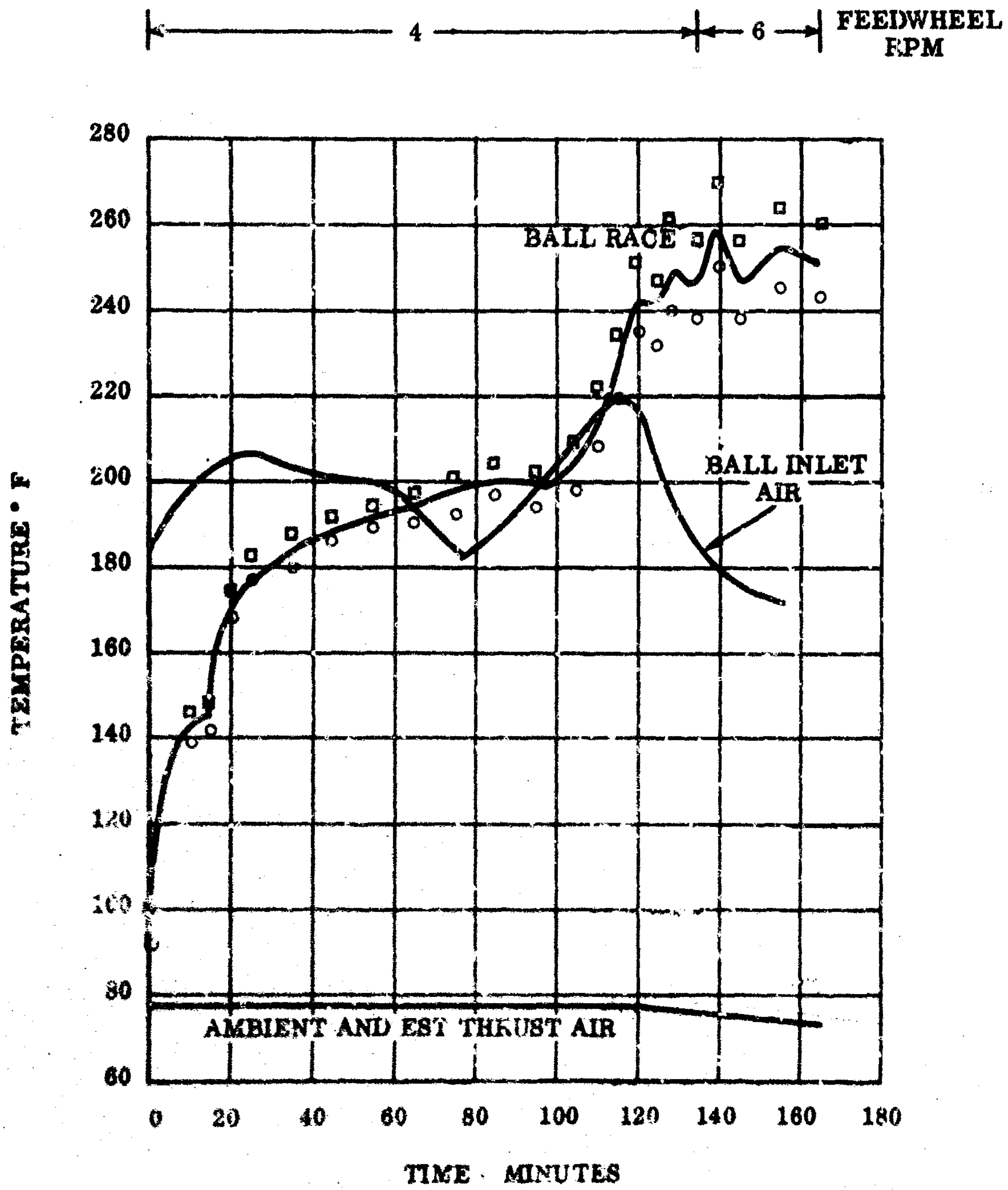

Figure 92. Engine Test F-85, Bearing Outer Race Temperature Versus Time

SPEED = 20,000 RPM
POWDER FLOW VARIED; FEEDWHEEL SPEED AS INDICATED; NORMAL IS 2 RPM
THRUST LOAD: 75 LB APPLIED AFTER 15 MINUTES

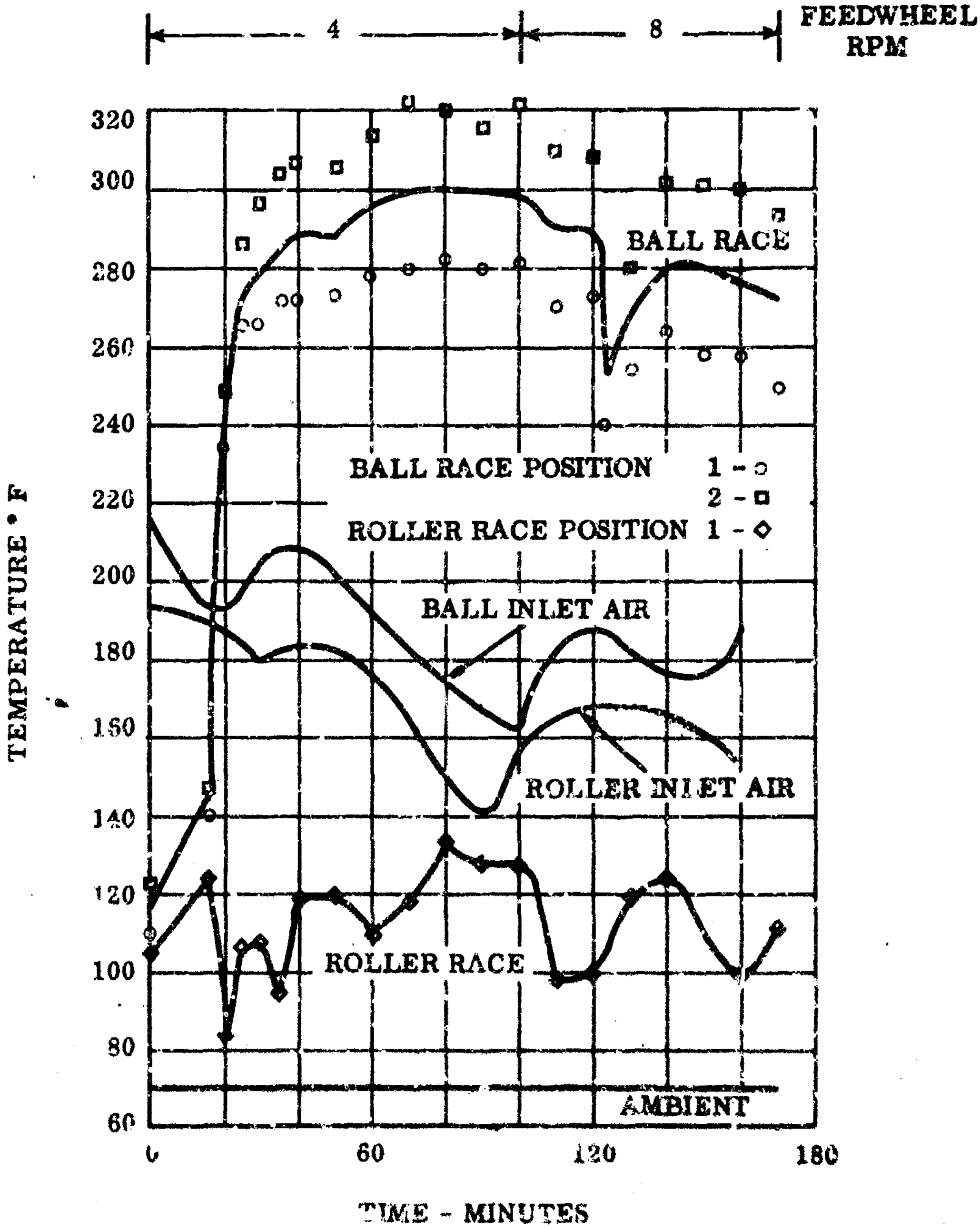

Figure 93. Engine Test E-86, Bearing Outer Race Temperature Versus Time

SPEED = 20,000 RPM
POWDER FLOW = 0.025 GRAM/MIN EACH BEARING (FEEDWHEEL 4 RPM)
THRUST LOAD: 100 LB APPLIED AFTER 15 MINUTES, REDUCED TO 88 LB BY 70 MINUTES
BALL RACE POSITION 1 - ○
2 - □
ROLLER RACE POSITION 1 - ◇

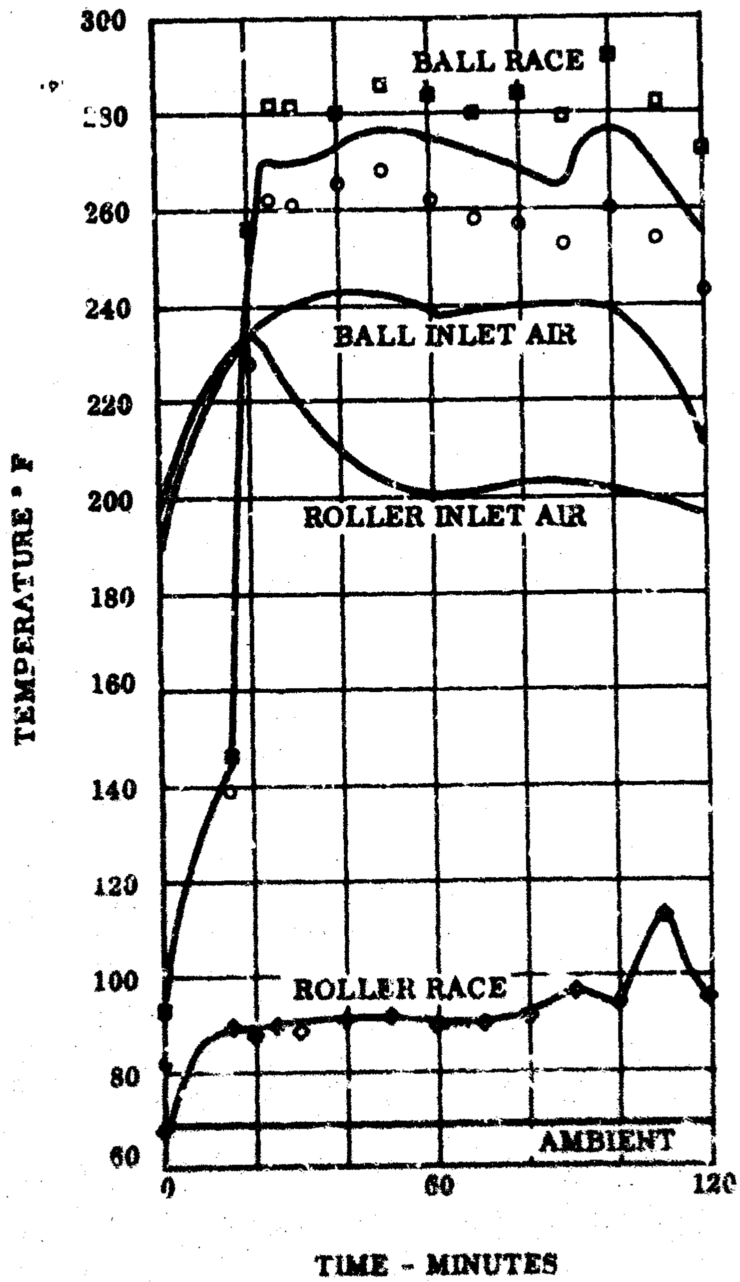

Figure 94. Engine Test E-67, Bearing Outer Race Temperature Versus Time

Unclassified

Security Classification

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(Security classification of title, body of abstract and indexing annotation must be entered when the overall report is classified)

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